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The diesel-electric pilot vessel 'Wyuna' of the Port Phillip Sea Pilots Association, Melbourne

The Diesel-Electric Pilot Vessel 'Wyuna'

By N. P. BLACKBURN, A.M.I.Mech.E., M.I.Mar.E., Manager, Marine Projects and Contracts

THE PILOT VESSEL 'WYUNA' completed successful trials in the Clyde during August 1953. This graceful yet practical ship was constructed to the highest classification of Lloyd's Register of Shipping for the Port Phillip Sea Pilots Association, Melbourne. The Australian aboriginal name 'Wyuna' means 'clear waters'. Mr. C. S. Mackinnon, Consultant Naval Architect of Melbourne, designed the vessel, which was built under his inspection by Messrs. Ferguson Brothers (Port Glasgow) Ltd. Her 'English Electric' twin screw diesel-electric propulsion machinery is of the latest type, and its design and construction illustrate this Company's wide experience in this field. The machinery was built to an arrangement agreed with the Naval Architect in the early stages of the vessel's design.

MAIN FEATURES

The leading particulars of the 'Wyuna' are:—

Length B.P.	185 ft 0 in.
Breadth, moulded	39 ft 0 in.
Depth, moulded, to main deck ..	15 ft 3 in.
Depth, moulded, to upper deck ..	22 ft 6 in.
Oil fuel bunker capacity	400 tons.
Fresh water capacity	85 tons.
Shaft horse power	1,400.
Designed speed	13 knots.

A pilot vessel is a special-purpose ship, in effect a floating station, from which pilots can attend ocean going vessels entering and leaving a port. Many ports do not require these vessels, it being possible to operate the pilot service from a station ashore. Working from such stations pilots keep 'watches' and when off duty they can return to their homes. The lot of pilots working from a pilot vessel is not so easy since they are confined to the ship for the whole of their duty spell, a period of two weeks in the case of the 'Wyuna's' staff.

The pilot vessel is a rendezvous where ocean going vessels make their first, or last, contact with a port. It must therefore restrict its movements to a limited orbit about its station, which is shown on the charts of the area. In fair weather the only drawback to this is the monotony but, in bad weather, conditions can be very arduous. The 'Wyuna's' station is at the seaward entrance to Port Phillip Bay, 45 miles from Melbourne; in this area very severe weather conditions arise at certain periods of the year.

These vessels do not, as a rule, lie at anchor at their station but hold their position by adjusting speed to match the flow of the tide. This results in long running periods at low power, but full power must be available at short notice to meet emergencies and to suit local conditions.

It is desirable that maintenance, refits and surveys be completed within the vessel's normal 'off station' period. The Port Phillip Sea Pilots Association operates two vessels each working a fourteen-day spell and thus, to maintain the relief roster, work must be completed in this time.

Fresh foodstuffs and consumable stores can be delivered to the vessel on station, and fresh water is at hand at her berth in Port Phillip. Fuel is not so readily available, and since bunkering involves a special trip to the fuelling station, incurring outlay in addition to the cost of the fuel, it is desirable to reduce such trips to a minimum.

In designing a pilot vessel consideration must be given to the foregoing points, and the 'Wyuna' is a very fine example of what can be achieved by close co-operation between the naval architect, shipbuilders and machinery suppliers. Since cost is a function of the size of the vessel she is as small as possible, compatible with carrying 20 pilots and a crew of 27. The accommodation is in excess of Ministry of Transport and Australian Navigation

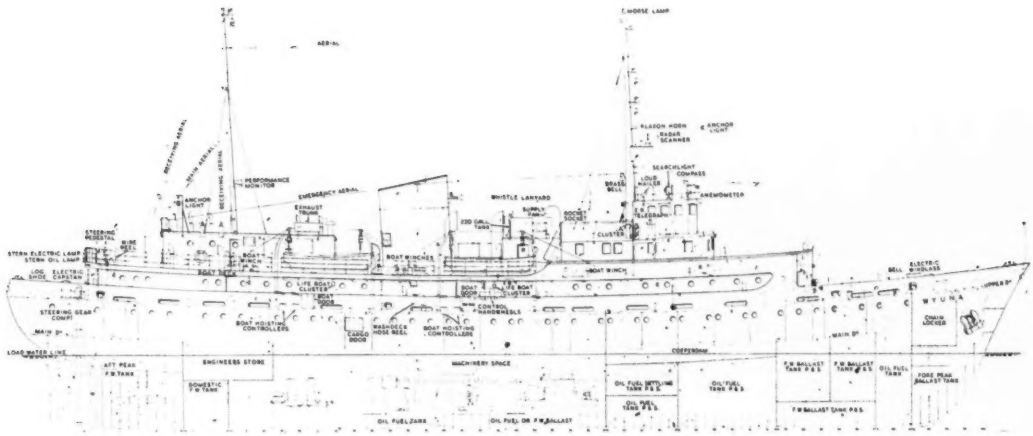

Fig. 1.—*General arrangement profile of the 'Wyuna'*

Act requirements, and the comfort of those aboard has been carefully considered. The compact arrangement of the vessel is shown in Fig. 1, whilst Figs. 2 and 3 give some indication of the high standard of the accommodation.

Separate dining rooms are provided for the pilots and ship's officers, and a spacious lounge is arranged on the boat deck. The pilots are accommodated in two-berth cabins under the bridge structure. The ship's officers are in single-berth cabins and each member of the crew has a single cabin. A recreation room for the crew and a spacious laundry with an electric washing machine add to the amenities. Fans supply fresh air which is distributed throughout the accommodation by directional louvres; heating is by electric radiators.

Fig. 2.—*Chief Pilot's cabin*

Noise and vibration are always unpleasant and are particularly so in a ship; care was taken to limit this source of annoyance in the 'Wyuna', with singularly successful results. Outside the engine room there is virtually no machinery noise and the vessel is almost free of vibration. The factors which contribute largely to this success are:—

The machinery ratings are moderate, and in the design of the engines particular attention is paid to noise level. Propellers, shafting and all rotating machinery are in perfect balance, and all generating sets are resiliently mounted,

Stanchions 'tying' the machinery space are insulated by rubber pads where they join the hull structure.

The engine room casing and deck head is clad with resin-bonded fibre glass, and air trunking is similarly treated after it leaves the machinery space.

Air velocity, at the various inlets and at the propulsion motor exhaust, is low.

The main engine exhausts are fitted with primary and secondary silencers.

Fig. 3.—Pilots' lounge

Since she has to remain on station in the worst weather experienced in the Bass Strait, with the minimum of discomfort to those aboard, the 'Wyuna' must be both 'sea-worthy' and 'sea-kindly'. Even to those unfamiliar with ships her hull, with its marked sheer, flared bow and high cruiser stern, promises ability to ride out the weather. The solid bulwarks and wide scuppers will give protection on deck and ensure the rapid shedding of any water which may come aboard.

Her inventory includes such aids as radar, echo-sounding gear, loud speaking telephones and loud-hailers, an air-speed indicator and a searchlight. With this equipment, and the up-to-date radio installation, she is well fitted for her duties.

She has to maintain for long periods a station-keeping speed as low as 1½ knots, and to meet the 'rip' which flows at the mouth of Port Phillip Bay, speeds up to 10 knots are required. As a maximum speed, to meet all contingencies, she has to be capable of 13 knots. In the design stage it was decided that the varying power requirements could be most satisfactorily met by installing diesel-electric machinery, with three main generating sets supplying power to the propulsion motors.

The use of diesel-electric machinery will reduce the time required for refits and surveys. Ordinarily the vessel's work can be done using two main generating sets only; this permits maintenance and survey to be carried out on the third set whilst the vessel is on station. Experience has shown that very little maintenance is necessary on the electrical machines and control gear in a diesel-electric installation, and such work can be readily carried out in the limited time available in port.

Careful attention was given to the hull with a view to extending its life and reducing maintenance. Cathodic equipment is fitted to protect the plating against corrosion, and it is anticipated that this protective equipment, by preventing the formation of rust, will not only increase the life of the hull plating but will also shorten the time necessary to clean down the hull prior to repainting. Since oil-bath stern tubes are fitted, sealed by patent glands, little wear should occur in the stern gear, and shaft withdrawals should be necessary only at infrequent intervals. These factors should ensure that the vessel spends the minimum time in dry dock.

The quantity of fuel carried by the 'Wyuna' is such that she need bunker only every six months. The ability to shut down main engines when on reduced power makes for economy, and the availability of auxiliary power from the main sets reduces the operating time of the independent auxiliary units, with resultant saving in fuel.

MACHINERY INSTALLATION

The 'English Electric' direct-current propulsion machinery delivers 1,400 s.h.p., as a continuous

tropical rating, to the propellers. It comprises three main generating sets supplying power to two propulsion motors. Fig. 4 shows the arrangement of the installation in the vessel.

The machinery was built to the requirements of Lloyd's Register of Shipping, for tropical service, and the insulation of the electrical machines and equipment is suitable for operation in a moist climate. The main generators are arranged in a 'series sandwich' circuit with the propulsion motors, and the plant operates on the 'controlled voltage' principle. The English Electric Company's standard, modified Ward-Leonard, system of twin screw control is employed; this permits one, two or three main generators to supply propulsion power, whilst retaining fully independent control of the propellers.

The system allows generators to be switched in or out of the propulsion circuit whilst the vessel is running, without interrupting propulsion power.

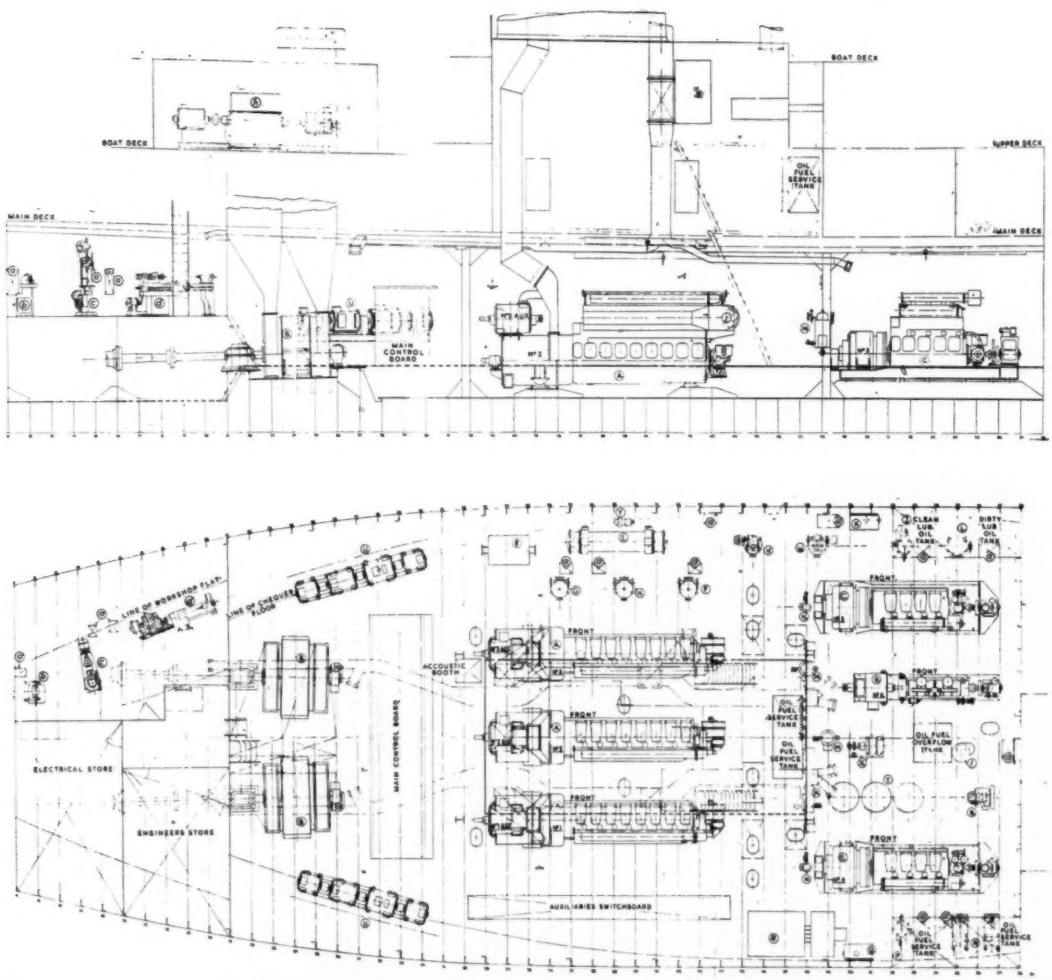

Fig. 4.—Plan and elevation of engine room, showing layout of machinery

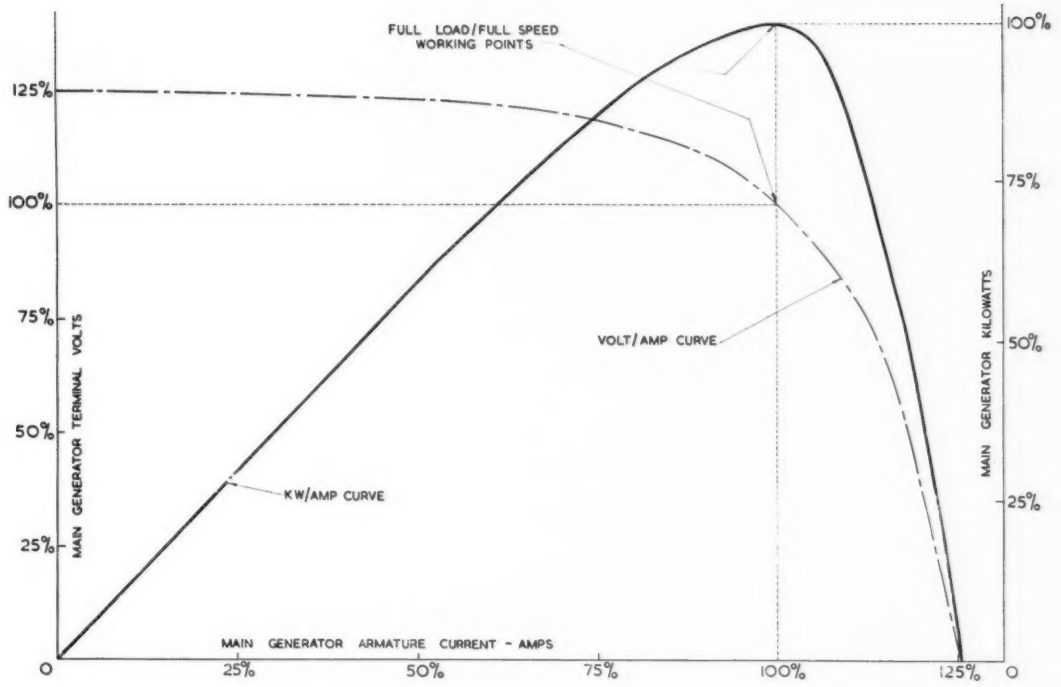

Fig. 5.—Main generator propulsion characteristics

The engines cannot be overloaded in any condition of operation or ' set up ', as the inherent characteristics of the system limit the power demand from the engines to their designed service rating.

Control System

In this control system the main generators are excited from a three-field exciter, the three separate field windings being:—

A self-excited shunt field connected through the propulsion controller resistors, and across the exciter terminals, and therefore responsive to the exciter voltage. The exciter voltage cannot build up on this field alone.

A separately excited field energised from the auxiliary busbars through the propulsion controller resistors. This field controls the build-up of the exciter voltage.

A negative series field carrying the main circuit current and opposing the self excited and separately excited fields.

The combined effect of these three fields is to give main generator characteristics as indicated in Fig. 5. The voltage is progressively reduced with increased current, and falls to zero at a predetermined point above the full load value. This results in a kW/current curve as shown, with maximum power occurring at the normal working point. The main generators therefore have inherent non-overloading characteristics and, with the equipment correctly adjusted, the service rating of the diesel engines cannot be exceeded, regardless of the number of generators in circuit or the position of the propulsion motor controllers. This characteristic also safeguards the machinery when the resistance on the propeller increases due to external conditions, but it does allow the production of higher than full load torques, up to a predetermined point of stall.

The characteristics of the main generators are maintained at lower speed settings, since movement of the propulsion controllers reduces the separately and self excited field currents of the 3-field main

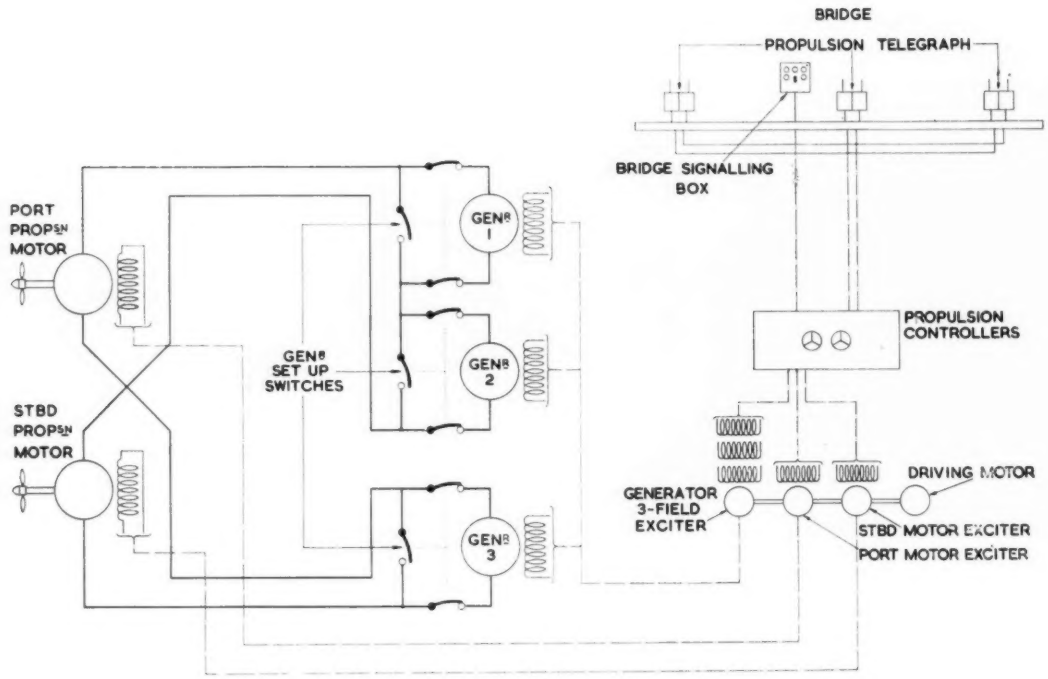

Fig. 6.—Simplified schematic diagram of electrical propulsion system

generator exciter. The separately excited field and self excited field of the main generator exciter (and therefore the voltage of the main generators) retain the same polarity with the propulsion controllers in the 'ahead' or 'astern' position, the change of direction of propulsion motor rotation being obtained by reversal of motor field polarity.

Each propulsion motor is separately excited from its own exciter. These exciters have separately excited fields, energised from the constant voltage auxiliary supply through potentiometers in the respective motor controllers. They have also negative differential fields in series with their armatures and the relative propulsion motor fields, to increase the rapidity of response to the controller movements. The propulsion motor controllers act on the self and separately excited fields of the generator exciter and on the field of the relevant propulsion motor exciter. The scheme in this installation is as shown on Fig. 6.

Propulsion Motors

Each of the two single-armature 600 volts D.C. propulsion motors (Fig. 7) delivers 700 s.h.p. at 150 r.p.m. They are totally enclosed machines, forced ventilated by a centrifugal fan which delivers filtered air from the fan room on the boat deck. The exhaust air is trunked to a 'mushroom' escape above the fan room. Rotatable brush gear is fitted, to simplify maintenance, and the motor frames are split at the horizontal centre line. Jacking equipment is provided to raise the armature shafts for the removal of bearing bushes. The endshields are fabricated and are built up in readily portable sections to give easy access to the commutator and back ends. The driving ends of the shafts are carried in 'Michell' combination thrust and journal bearings, and 'Michell' journal bearings support the free ends. Revolution counters are fitted. Dial thermometers record the temperature of the exhaust air, and they also operate a 'high temperature'

alarm in the event of failure of the ventilating air supply.

Control Gear

All switchgear and control equipment related to the propulsion machinery is contained in the main control board which is installed athwartships between the main generating sets and the propulsion motors. This is a 'dead front' sheet steel drip-proof cubicle, canopy protected and louvre ventilated, with hinged removable doors at the back and hinged panels at the front permitting easy access to all components. Fig. 8 shows the external appearance of the board and the comprehensive instrumentation which allows for rapid monitoring of the performance of the machinery.

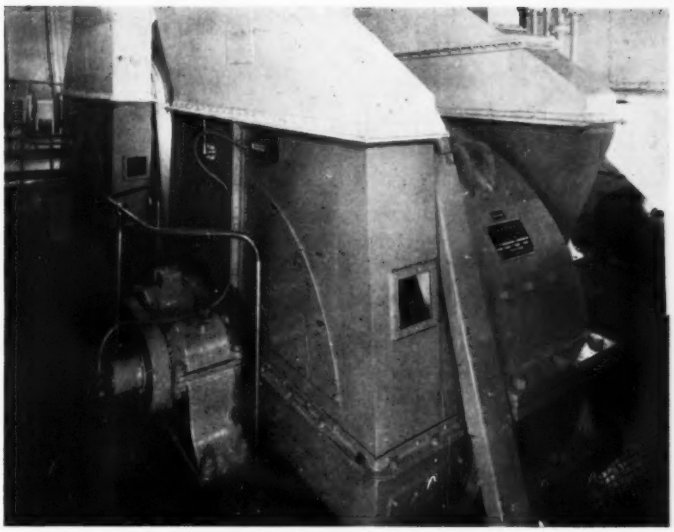

Fig. 7.—The two 700 s.h.p. 600 volts D.C. propulsion motors

The section of the control board on the left of the illustration contains the three main generator 'set up' switches, field switches and the voltmeters and kilowatt meters which register the output from each set. The centre section contains the propulsion motor controllers, the bridge signalling switch, indicating lamps and supervisory instruments. These instruments include an s.h.p. meter and revolution indicator for each propulsion motor and an ammeter which shows the main circuit current. The excitation section is on the right and contains the circuit breaker, changeover switch and exciter set starters.

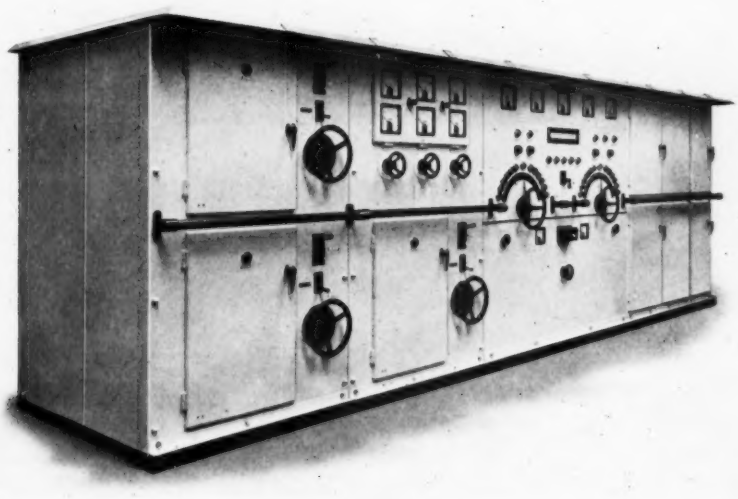

Fig. 8.—Main propulsion control board

The propulsion controllers have six speed steps ahead and astern; they are motorised for remote control from the bridge telegraphs (Fig. 9) and, alternatively, they can be operated manually at the control board. Any order transmitted by the bridge telegraph registers on one of the lights disposed around the controllers: automatic or manual movement of a

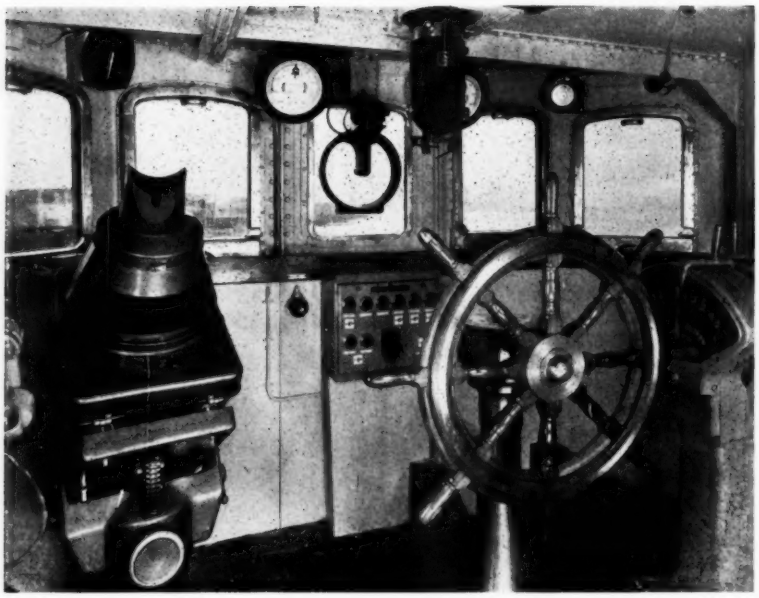

Fig. 9.—The wheelhouse, showing one of the telegraph pedestals, the control panel and the port and starboard shaft speed indicators

controller to the position indicated by the light adjusts its propulsion motor and automatically acknowledges the order to the bridge. Propeller speeds between the six speed steps on the controller are obtained by adjusting the main generator rheostats. The propulsion motor ventilating fan starts automatically when either propulsion motor controller moves beyond the second speed step, and stops when both motor controllers are in the 'stop' position. Failure of the ventilating air supply does not immobilise the vessel since the design of the propulsion motors permits them to take currents up to 670 amperes on two sets and 580 amperes on three sets, without the air supply.

A control panel on the bridge permits the transfer of such orders as 'standby', 'finished with engines' and 'take over control', and contains indicating lights showing the number of generating sets on propulsion. These orders are acknowledged by indicating lights which are energised by the signalling switch on the main control board. Propeller-speed indicators on the bridge repeat the readings from the main control board and are marked to show the speed range available with one, two or three generating sets in operation.

Generating Sets

Each of the three main generating sets (Fig. 10) consists of an 'English Electric' type 8SRKM diesel engine, developing 635 b.h.p. at 700 r.p.m., driving a 380 kW 400-volt main generator and a 60 kW 220-volt D.C. auxiliary generator at constant speed. The main generator is direct coupled and flange mounted to the engine, whilst the auxiliary generator is mounted on top of the main generator and chain driven from it. The complete unit is mounted on the 'English Electric' system of three point support, and Dexine pads insulate the mountings from the ship's structure. This arrangement obviates the need for a combination baseplate, gives free access about the engine and generator, and protects the set from malalignment should the vessel 'work' in a seaway.

Both the main and auxiliary generators are self ventilated by integral fans mounted on their armature shafts; air is trunked into the machines from louvres at the back of the funnel and is exhausted into the machinery space. The machines are totally enclosed and are fitted with marine-type journal bearings, self lubricated from integral oil wells. The triplex chain drive to the auxiliary generator is

enclosed in a fabricated steel casing and is lubricated from the main engine system. Both machines are arranged for easy access to commutators, brush gear and windings, and inspection windows are fitted at the commutator end. The 60 kW constant-voltage generators are designed to operate in parallel with each other, and with the two 150 kW independent auxiliaries and the 50 kW port set (described later under 'Auxiliaries'); this permits the most economical combination to be selected to meet the demand for auxiliary power.

Engines

The 8SRKM engine (Fig. 11) is the marine version of the 'English Electric' standard eight-cylinder, in-line, four stroke, exhaust turbo-charged diesel engine. Its cylinders are 10″ bore × 12″ stroke, it has 'four valve' heads, and develops 900 b.h.p. at 750 r.p.m. (12-hour B.S.I. rating). Lubrication, on the semi-dry sump principle, is provided by a gear-type pump driven from the engine crankshaft. This pump draws from the engine sump tank, located in the double bottom, and delivers through an oil cooler and filters to the main supply rail. After passing through the engine the oil returns by gravity to the sump tank. A 'Streamline' filter, operating on continuous by-pass, is incorporated in the lubrication system of each engine, and a motor driven lubricating oil pump is provided for standby service to any of the three sets.

The engines are fresh-water cooled and the circulating water temperature is automatically maintained at the correct level by thermostatic control. Three independent motor driven pumps, one for fresh water, one for salt water and one stand-by to both, operate the main engine cooling system, and the jacket water from all three engines is cooled by a single heat exchanger (Fig. 12). Before entering the heat exchanger the hot water from the engine jackets passes through a calorifier which heats the ship's domestic hot water supply. Electric heaters are fitted in the calorifier for use when the main engines are stopped.

Auxiliaries

Two 150 kW diesel generating sets are installed,

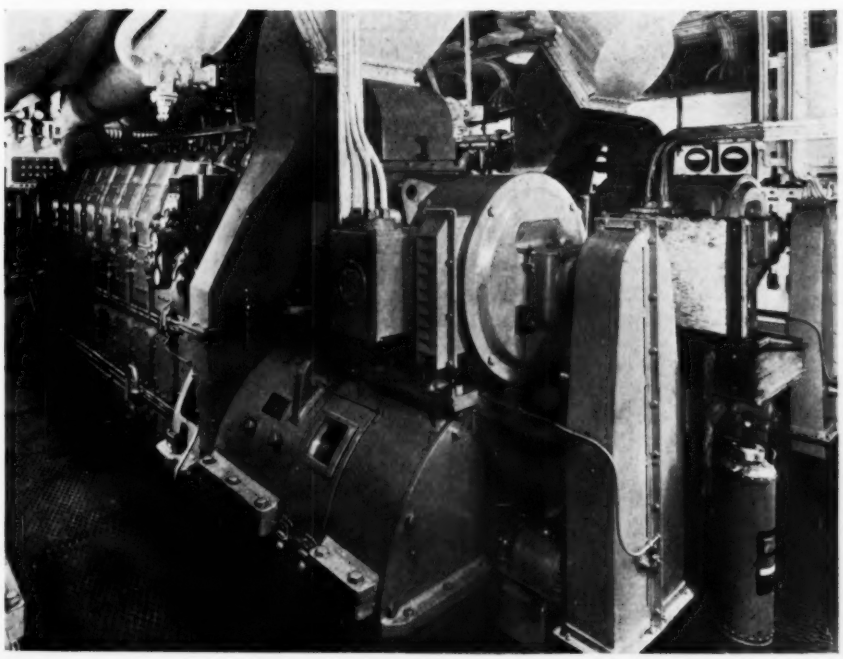

Fig. 10.
The centre main generating set, looking forward

forward of the main generating sets. These units each comprise an ' English Electric ' type 4RKM engine direct coupled to a 700 r.p.m., 220 volts D.C., level compound wound, drip-proof generator; a 20 c.f.m. air compressor is clutch-coupled to the pump end of the engine. The engine, generator and compressor are carried on a resiliently mounted combination baseplate. Fresh-water cooling is employed and a heat exchanger is provided for each unit; circulation is by engine-driven fresh and salt water pumps. The lubrication arrangement is the same as employed on the main engines, but no standby lubricating oil pump is included.

These auxiliary engines are from the same range as the main engines, and have the same bore and stroke. They are four cylinder, four stroke, naturally aspirated units with ' four valve ' heads, and develop 300 b.h.p. at 750 r.p.m. as their 12-hour B.S.I. rating. To attain the perfect balance necessary to ensure freedom from vibration, they are fitted with Lanchester balancer gear. Being from the same range as the main engines the working parts are largely interchangeable and service requirements can, in the main, be met from a common stock of spares. This provides economy in capital cost and storage space, amongst other advantages.

A 50 kW port set is installed between the two 150 kW auxiliary sets and comprises an ' R.N.' six cylinder engine direct-coupled to an ' English Electric ' 1,000 r.p.m. 220 volts D.C. generator. A 28.5 c.f.m. air compressor is clutch-coupled to the pump end of the engine and the whole assembly is carried on a resiliently mounted combination baseplate. The set has self contained lubricating oil and salt-water/fresh-water systems and is arranged for hand starting if necessary. Being independent of the rest of the installation this set can be used to start up the plant when no services are otherwise available. Its capacity is sufficient to meet all domestic power, ventilation and lighting requirements in port, when the crew only are aboard, with a reserve to permit essential engine room services to be maintained.

For economic reasons, and to relieve the engine room staff of watchkeeping duties in port, it is very desirable to be able to shut down the auxiliary engines when berthed alongside. To enable this to be done without affecting the comfort of those living aboard, the ' Wyuna ' can take a shore supply. The 415 volts 3-phase supply is reduced to 220

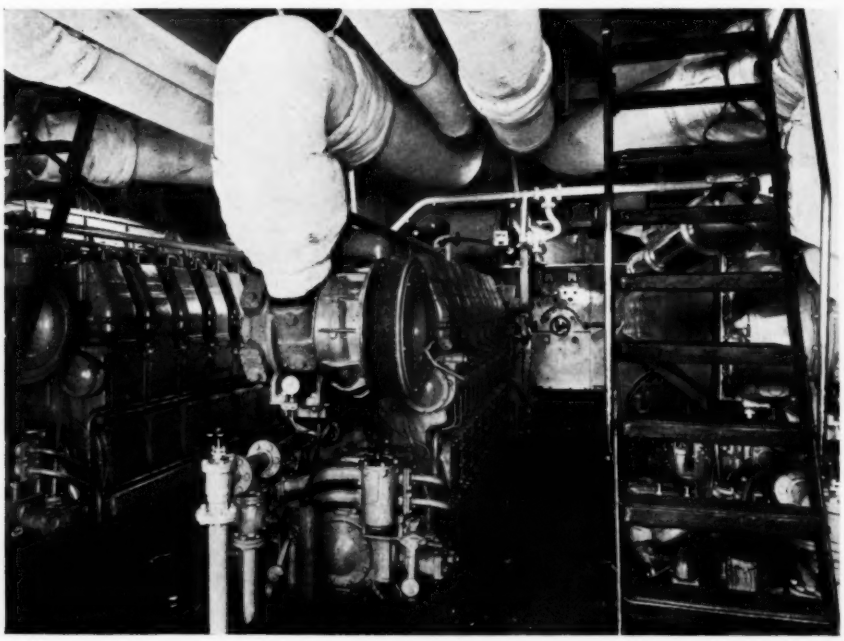

Fig. 11.
The engine room, looking aft, showing the three main generating sets

volts by a transformer and fed to a mercury-arc rectifier with an output of 160 amp at 220 volts D.C. This supply, and that from the constant voltage auxiliary generators, is controlled and distributed from the 'dead front' auxiliary switchboard on the starboard side of the engine room.

Apart from the alarms and fault indication equipment built into the control gear as protection on the electrical side, such protection is extended to other parts of the installation. Low pressure and high temperature alarms are fitted in the lubricating oil and cooling water circuits of the main and auxiliary engines, and high/low level alarms are fitted in the oil fuel settling and daily service tanks. A panel, at the engine room entrance from the main deck, carries capacity gauges for fuel settling and bunker tanks; these indicate the condition of all fuel tanks at a glance.

The remainder of the engine room auxiliaries comprise, an 80-ton ballast pump, a 50-ton bilge pump, two 7-ton oil fuel transfer pumps and a 5-ton domestic fresh water pump. Two centrifugal separators for fuel and lubricating oil are fitted, each equipped with electric heaters. Compressed air is stored in four 23 cu ft capacity air receivers, and a 6 c.f.m. diesel driven air compressor is installed as reserve to the compressors driven from the auxiliary generating sets. The electrically driven auxiliaries are powered by ' English Electric ' marine-type motors. A useful workshop is situated in the port side of the engine room aft and is equipped with a lathe, a pillar-type drilling machine and a tool grinding machine, in addition to the usual vice bench and tool lockers.

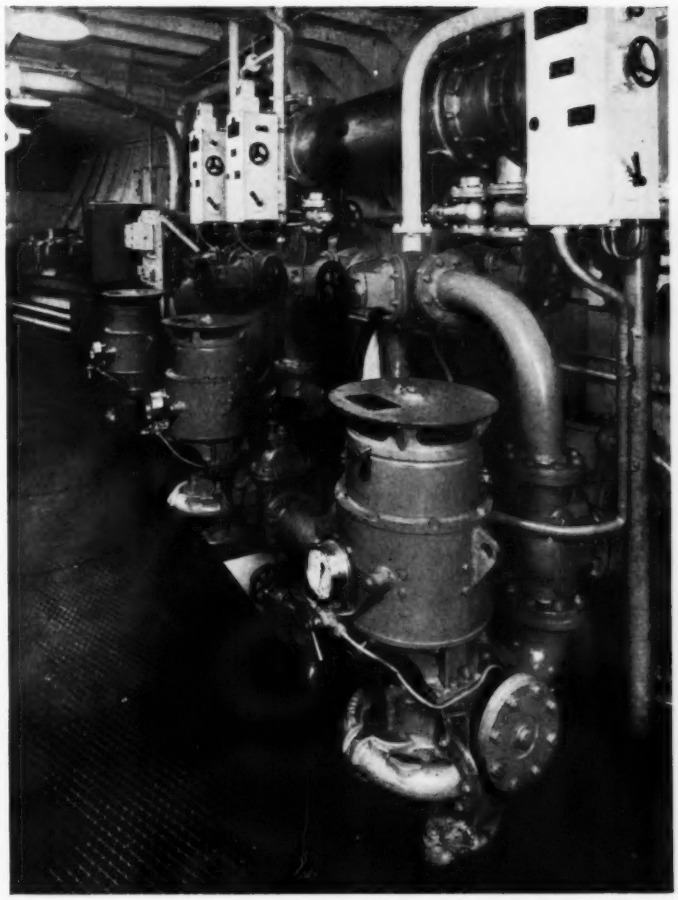

Fig. 12.—Cooling water circulating pumps and heat exchanger, with exciter set in background

Engine Room Ventilation

A well ventilated engine room is necessary if the efficiency of the personnel is to be maintained, and in this respect the installation in the ' Wyuna ' is very successful. A direct air supply of 10,000 c.f.m. is provided by two motor driven fans; this air is distributed about the machinery space, above the working platform level, by trunking. The supplementary supply exhausted from the main generators also approximates to this quantity when all three sets are running, and is distributed over the tank tops beneath the working platform. The direct supply prevents the formation of hot ' pockets ' whilst the supplementary supply, in escaping past the edges of the platform in way of the machinery items, drives the generated heat and fumes upwards to the top of the engine room. The funnel casing is

open to this space, and apertures at the funnel top act as a ventilation exhaust.

This system, which is aided by the natural tendency of hot air to rise, is further assisted by the updraught caused by the heat radiated from the silencers, which are located in the funnel.

On trials, with the machinery at full power, the maximum temperature at working platform level was only 10°F above the ambient temperature. Further, and of equal importance to temperature, the atmosphere was completely free of the fumes and oil vapour which so frequently cause discomfort in the smaller diesel engine rooms.

A measure of control is available in the system since, when the ambient temperature is low, one or both of the direct supply fans can be stopped. The supplementary supply is reduced when main generating sets are shut down but, as this ' set up ' condition is synonymous with a reduction in power, the heat and fumes generated are reduced proportionately.

TRIALS

It has been mentioned that trials were successful; the performance of the vessel and her machinery was most satisfactory. With the machinery operating at its service full power the vessel attained a speed of 13½ knots, a comfortable margin on the designed 13 knots. Similarly the speeds of 9½ and 11½ knots, reached on one and two main generating sets respectively, were just above the estimated figures.

Before trials the propulsion motor controllers were set to move from full power ' ahead ' to full power ' astern ' in 20 seconds. It was not considered necessary to alter this adjustment, since from running ahead at 13½ knots the vessel began to move astern within 80 seconds on this timing. Conversely, from full speed astern, movement ahead commenced within 50 seconds. These timings mean that the vessel moves approximately 250 yards, after the telegraph is moved, before coming to rest; this is the equivalent of four ship lengths.

The ' Wyuna ' made the voyage from Glasgow to Melbourne under her own power, arriving there on October 13th 1953; she went into service in November that year after a post voyage docking.

The Napier 'Nomad' Compound Diesel Aero Engine

The article in the September 1953 *issue of this Journal, Volume* 13, *No.* 3, *gave particulars of the performance, economic aspects and operational features of the 'Nomad' engine compared with aero engines of other types. The present article describes the 'Nomad' engine itself.*

THE NAPIER 'NOMAD' is the first aero engine in the world designed *ab initio* to exploit the unique advantages to be gained from associating a piston engine and a gas turbine engine to form a compound power unit.

By suitably manipulating the cycle of operations the most valuable features of both of these major components have been incorporated in the design, which has resulted in a prime mover possessing outstandingly low fuel consumptions which are maintained unimpaired over a wide range of powers, altitudes and flight speeds. In addition, the engine is capable of operating efficiently with diesel fuel which, because of its low price, enhances still further the already low fuel cost figures. The engine will operate equally well on kerosene or wide-cut gasoline.

These factors make the engine particularly suitable for the rôle of maritime reconnaissance in the military field and for air freighting operations in the civil field. It is becoming generally recognised that we are approaching a period which will show a tremendous increase in air freighting. The Nomad engine may well provide the solution to the problem of really economic air freight charges, where economy of operation is of the utmost importance.

Leading Technical Data (N. Nm. 6. Rating)

Performance at sea level :

Take-off ..	3,046 s.h.p. at .355 lb /s.h.p.-hr.
(2,050 r.p.m.)..	3,135 e.h.p. at .345 lb/e.h.p.-hr.
Max. continuous	2,392 s.h.p. at .358 lb /s.h.p.-hr.
(1,900 r.p.m.) ..	2,448 e.h.p. at .350 lb /e.h.p.-hr.
Cruising ..	1,849 s.h.p. at .359 lb /s.h.p.-hr.
(1,750 r.p.m.) ..	1,893 e.h.p. at .351 lb /e.h.p.-hr.

Performance at altitude (I.C.A.N. atmosphere 300 knots T.A.S.) :

Operational necessity at 11,000 feet ..	3,110 s.h.p. at .346 lb /s.h.p.-hr.
(2,050 r.p.m.) ..	3,250 e.h.p. at .333 lb /e.h.p.-hr.
Max. continuous at 19,000 feet	2,480 s.h.p. at .346 lb /s.h.p.-hr.
(1,900 r.p.m.) ..	2,584 e.h.p. at .332 lb /e.h.p.-hr.
Cruising at 25,000 feet ..	1,952 s.h.p. at .340 lb /s.h.p.-hr.
(1,750 r.p.m.) ..	2,024 e.h.p. at .327 lb /e.h.p.-hr.

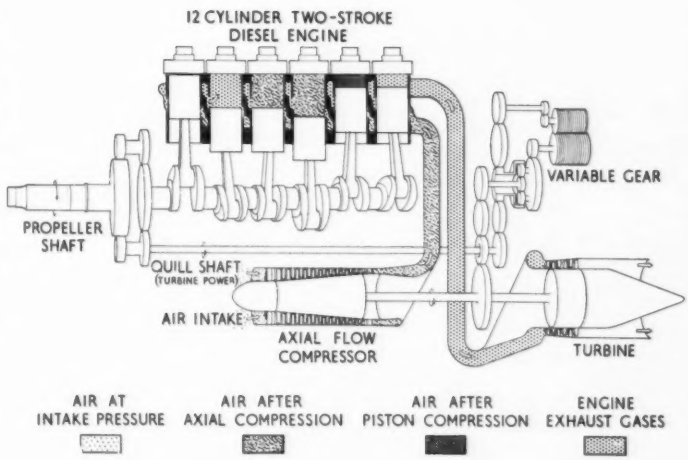

Fig. 1.—*Diagram of Nomad engine*

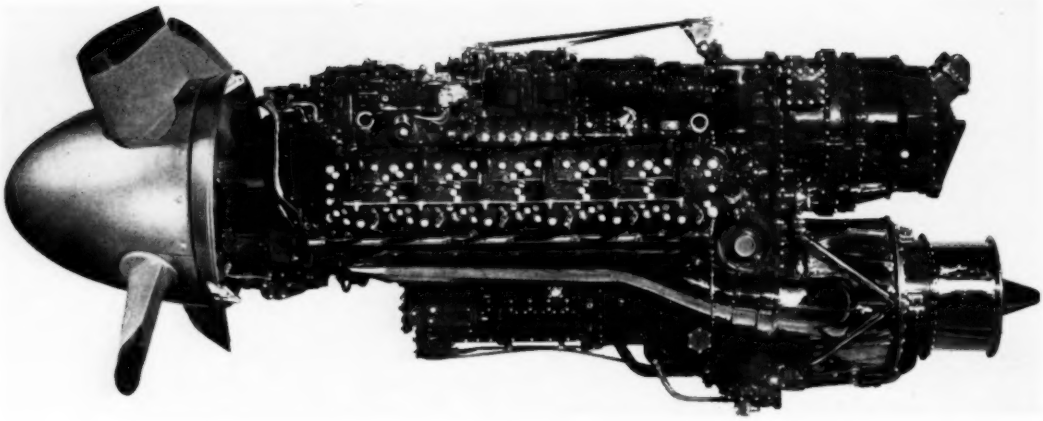

Fig. 2.—Side view of engine

Net dry weight : 3,580 lb.

Propeller types :

Rotol (duralumin blades) or De Havilland (hollow steel blades). — In sizes from 13 ft to 16 ft diameter.

Reduction gear ratios :

.526 : 1, .555 : 1, .569 : 1, .614 : 1, .660 : 1.

Fuels :

Wide-cut gasoline to Specn. D.Eng.R.D. 2486 or kerosene to Specn. D.Eng.R.D. 2482 or diesel fuel, Pool Gas Oil.

Lubricating oil : to Specn. D.Eng.R.D. 2472B/2.

Mechanical Layout

The mechanical layout of the Nomad engine is shown in diagrammatic form in Fig. 1, from which it will be seen that the two main components, the diesel engine and the turbo-compressor set, are connected by a gear train incorporating an

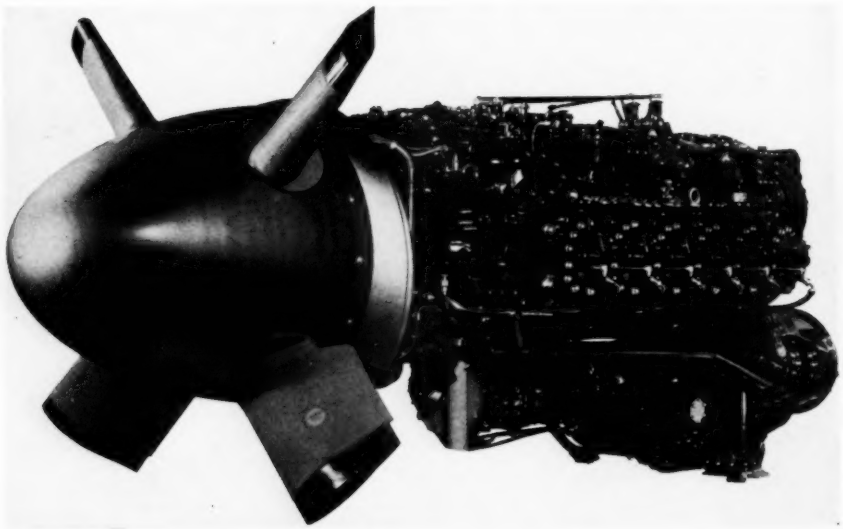

Fig. 3. Three-quarter front view of engine

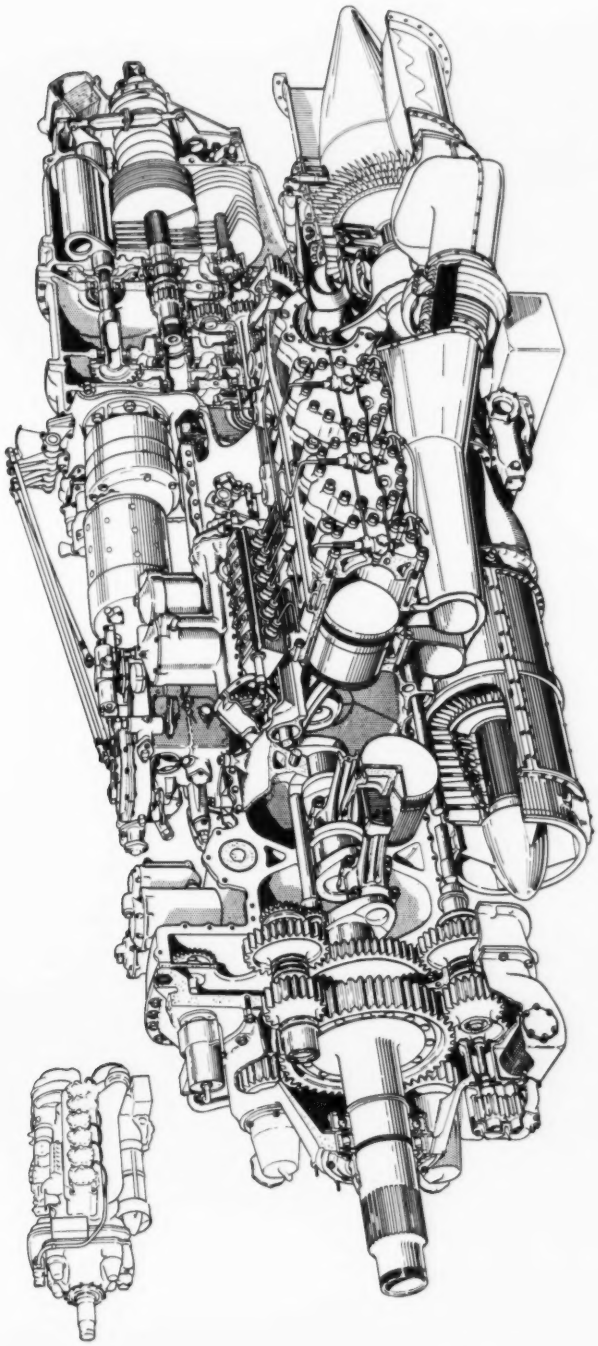

Fig. 4.—'Cut away' view of engine

' infinitely variable ' gear, which enables the selected boost to be maintained over a wide range of altitudes. External views of the engine are shown in Figs. 2 and 3, and Fig. 4 shows a ' cut away ' view of the engine.

The Diesel Engine

This comprises a 12-cylinder horizontally-opposed unit operating on the two-stroke cycle, the cylinder bore and stroke being 6 in and $7\frac{3}{8}$ in respectively, giving a total displacement of 41.1 litres.

The crankcase is of magnesium-zirconium alloy and is split on its vertical centre-line to permit assembly of the six-throw nitrided steel crankshaft which is carried in thin-wall bearings of lead-bronze type.

The two cylinder blocks are aluminium-alloy castings and contain water-jackets for cylinder cooling. The dry cylinder liners, which are of copper-chromium alloy, contain the inlet and exhaust ports which are piston operated, there being no valve gear of any kind. The scavenging system employed, which is illustrated in Fig. 5, gives a well-defined air movement within the cylinder and combines effective scavenging of burned gases from the cylinder with minimum pressure losses.

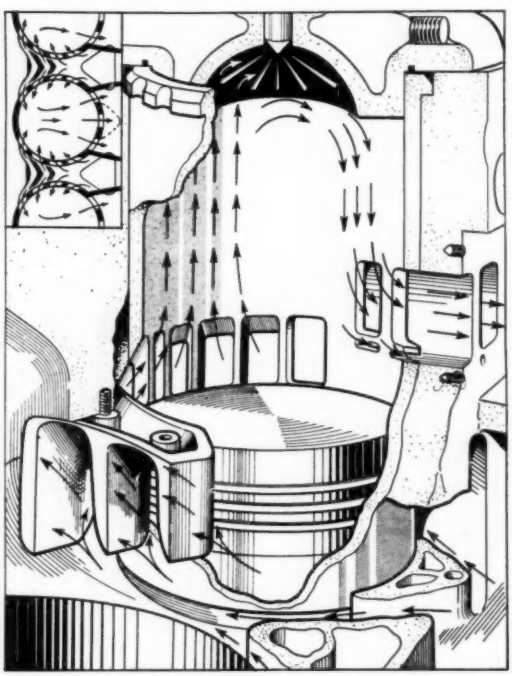

Fig. 5.—Arrangement of engine cylinder

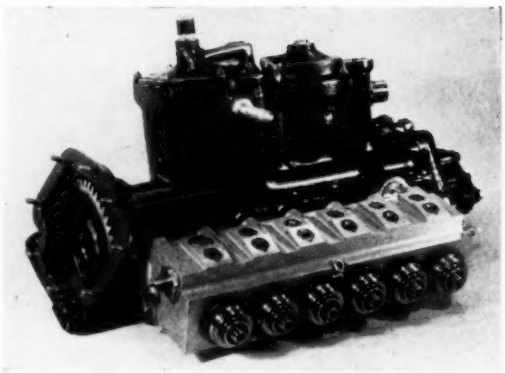

Fig. 6.—Fuel injection pumps

The combustion chamber is formed in the cylinder head and is hemispherical in shape. It has a centrally situated fuel injector having one central spray orifice and five equally spaced radial orifices, the radial sprays being directed at the combustion chamber walls. The injector is of special design to suit the requirements of high-speed operation and high cylinder outputs.

The injection pumps are mounted in two blocks of six elements on the port and starboard flange positions on top of the crankcase. They are of ' jerk-type ' design, specially developed for operating at high speeds and specific outputs ; an exterior view is shown in Fig. 6.

The pistons (Fig. 7) are of two-piece construction, comprising a Y alloy body with an upper portion in austenitic steel, which permits the latter to run at temperatures of upwards of 600° C at the centre of the crown at full power. Cooling oil, pressure fed from the crankshaft system through drillings in the connecting rod, is provided and is directed particularly at the region behind the piston gas rings to ensure that the ring temperature is reduced to a satisfactory working level.

Advantage has been taken of the substantially uni-directional loading on all connecting rods,

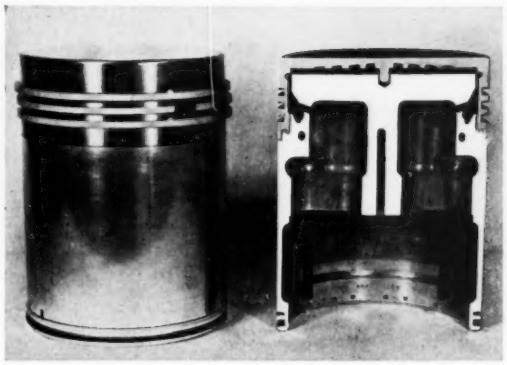

Fig. 7.—Arrangement of piston

which results from operating on a two-stroke cycle, to use unconventional forms of big end and small end bearings—this being demonstrated in Fig. 8. It will be seen that both bearings are of the half-bearing or ' slipper ' type, with straps fitted as safety devices to deal with the light loads in the reverse loading direction.

Further details of the cylinder and connecting rod arrangements are shown in Fig. 9.

The Axial-flow Compressor

The axial-flow compressor is suspended by four flexible links below the crankcase, as shown in Fig. 10, and its entry is arranged to take full advantage of ' ram ' air.

It has 12 stages and a maximum pressure ratio of $8\frac{1}{4}$: 1 with an air mass flow of 13 lb per second. Adjustable inlet guide vanes are employed to extend the operating range of the compressor at low speeds to match the requirements of the turbine, and these are automatically controlled.

The rotor and stator blades of the first 5 stages are steel, those of the remaining stages being aluminium-bronze.

The Turbine

The turbine, which is mounted co-axially with the compressor, has three stages and is designed to extract the maximum energy from the gases ; thus there is only a small amount of residual thrust. It is attached to the rear gear casing of the engine by a tubular framework, and both the turbine and compressor are connected into this casing by shafts having internally-toothed couplings at each end to deal with any slight expansion.

Rear Gear Casing

The rear gear casing is machined from a magnesium-zirconium casting and contains the main portion of the reduction gear train between the turbine and the diesel engine. In addition, it houses the oil sump, the oil pressure and scavenge pumps, and the coolant pumps, and contains drives for the engine starter and auxiliary gearbox.

The Infinitely Variable Gear

This is contained in a housing which is attached to the rear face of the rear gear casing above the turbine. This gear has been designed and developed

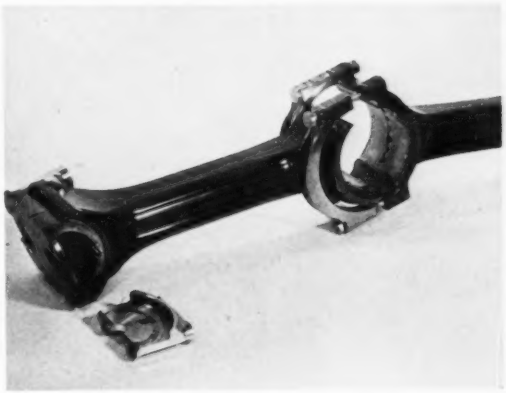

Fig. 8.—Arrangement of connecting rod

by D. Napier & Son Ltd. to suit the particular requirements of the Nomad application, but the basic scheme was evolved by Dr. Josef Beier and developed in Germany before the last war. It is an ingenious practical adaptation of the well-known scheme in which two conical members are end loaded together, the gear ratio being varied by sliding one cone over the other.

Details of the variable gear are shown in Fig. 11. A ' pack ' of discs with narrow conical rims are arranged on a central shaft and are spring-loaded to trap a series of coned discs carried on three planetary shafts disposed around the central shaft. Each planetary shaft swings around a fulcrum to obtain changes in gear ratio, and by attaching a gear to the planetary shaft meshing with one at

the fulcrum it is possible to arrange a constant mesh gear system so that changes in gear ratio can be made while running.

In operation, the gear is lubricated by high-pressure oil directed at the contact points between each flanged and coned disc. This prevents metallic contact between them, the power being transmitted through the drag force obtained by fluid shear of the oil film.

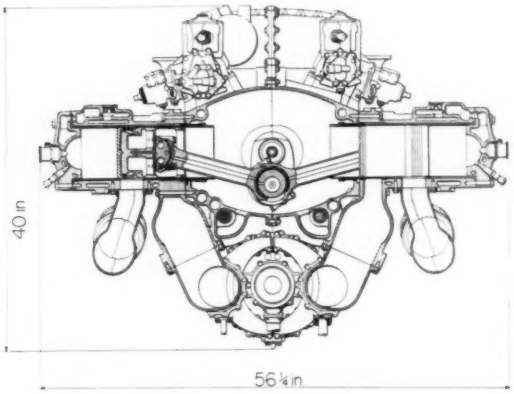

Fig. 9.—Cross-section through engine

To reduce the dimensions and weight of the variable gear to a minimum, it is connected into the gear system of the Nomad as indicated in Fig. 12.

Here the speed reducing gear for the turbo-compressor set is of epicyclic form, the variable speed device being connected across two members of the epicyclic train. This, in effect, produces two mechanical systems in parallel, only about 30% if the total power being transmitted through that part of the system in which the speed varying device is fitted.

Engine Controls

The engine is fitted with a single lever control system which relates the speed, boost, and fuel functions to ensure that a pre-determined operating line is followed. The speed control is provided by a mechanically set constant speed unit varying the propeller pitch. For each speed selected there is a fixed value for boost pressure and for fuel flow. The boost pressure obtained is applied to a normal type of variable datum boost control unit, the servo-piston of which is connected by linkage with the control setting of the variable gear. By this means the gear ratio is automatically adjusted to

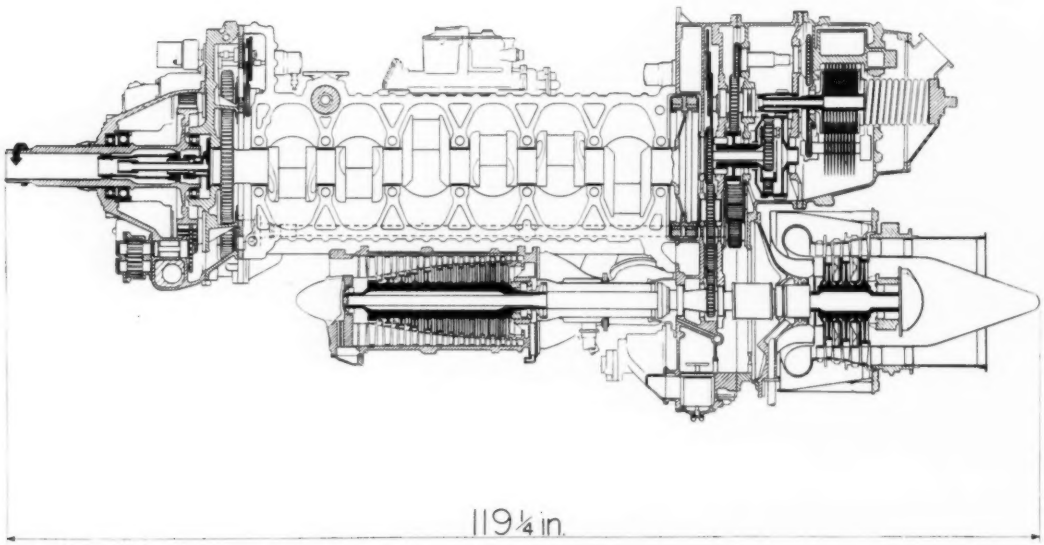

Fig. 10.—Longitudinal section through engine

speed up the compressor to maintain the selected boost pressure with increasing altitude until the maximum permitted r.p.m. of the turbo-compressor set is reached. This corresponds to the maximum power altitude for that particular engine condition, and above this point the fuel flow is reduced automatically by a re-setting device in sympathy with falling boost.

Thus the engine is capable of maintaining substantially constant power over a wide range of altitudes. For example, under cruising conditions the operating range over which these conditions apply extends from sea level up to 25,000 feet.

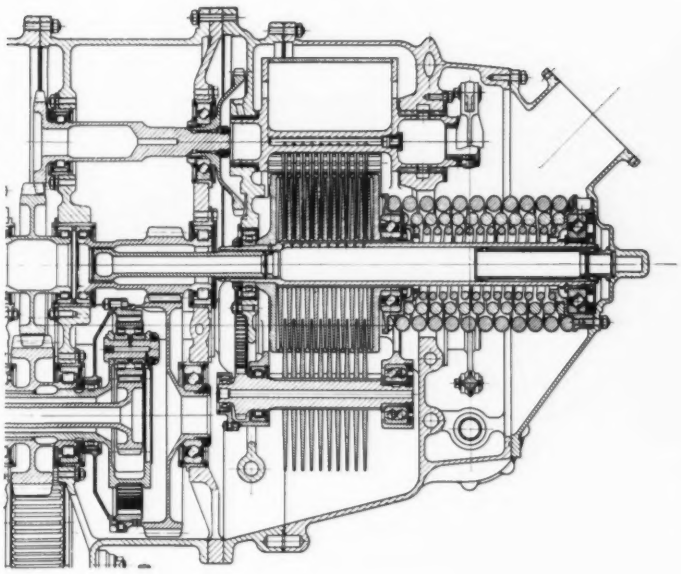

Fig. 12.—Variable gear as fitted to engine

Starting

Starting is by means of a 112-volt electric starter which is geared into the crankshaft system. As the compression ratio of the diesel engine is only 8 : 1 it is necessary to fit a sparking plug in each cylinder, for use when starting only, energised by a high-energy ignition system of Napier design.

Fig. 11.—Three-quarter view of variable gear

Engine Performance

The exceptionally low fuel consumption of the Nomad engine is a direct result of the use of a compound cycle which enables very much higher overall compression and expansion ratios to be employed than are possible with either petrol engines or gas turbine engines.

For example, under take-off conditions at sea level, the effective overall compression ratio is 27 : 1 and the overall expansion ratio is 27 : 1, the boost pressure being 89 p.s.i. (absolute) and the brake mean effective pressure 205 p.s.i. The take-off power is 3,135 e.h.p. or 76 h.p. per litre, corresponding to a specific weight of 1.17 lb per h.p. The fuel consumption based on equivalent power is .345 lb per e.h.p.-hr.

With increasing altitude up to rated height the overall expansion ratio of the cycle increases in sympathy and this leads to an improvement in specific fuel consumption, a typical example being at the recommended cruising rating where a figure of .327 lb per e.h.p.-hr is obtained at 25,000 ft. Performance curves at various engine ratings are shown in Figs. 13 and 14 which relate to shaft

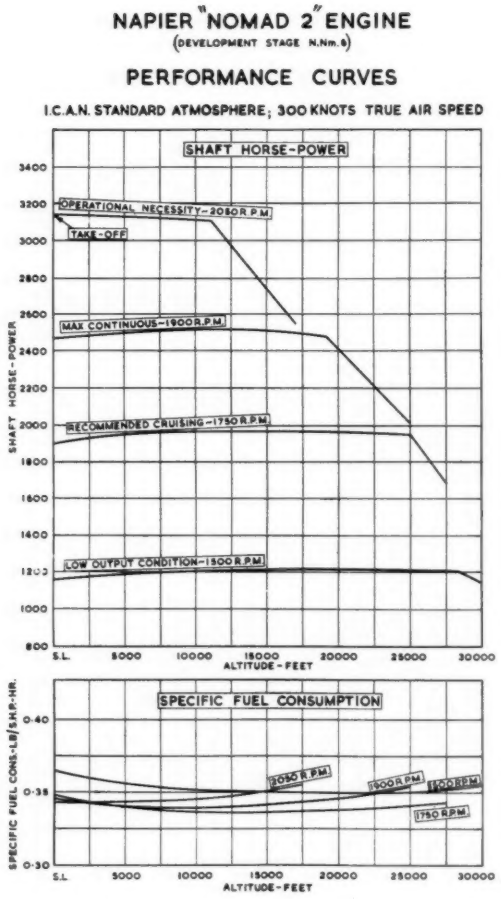

Fig. 13.—*Performance curves* (*s.h.p.*)

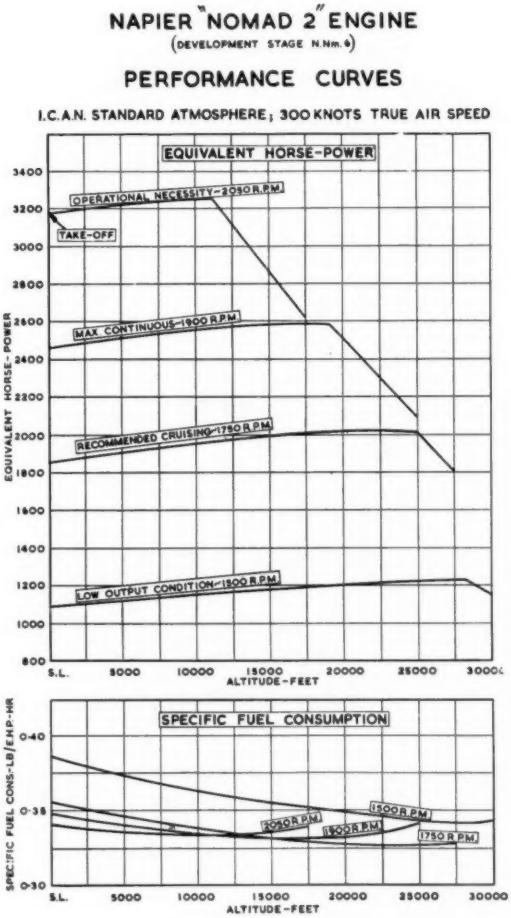

Fig. 14.—*Performance curves* (*e.h.p.*)

horse power and equivalent horse power respectively.

By using water injection into the engine manifolds, the take-off power of the engine is increased to 3,580 e.h.p. corresponding to 87.2 h.p. per litre and a specific weight of 1 lb per e.h.p.

Still further increases in take-off power can be obtained by burning a small quantity of additional fuel in the exhaust manifold between the engine and the turbine, there being sufficient excess air in the exhaust gases for this to be a practical proposition. By a combination of water injection and auxiliary combustion, as outlined above, the take-off power can be increased to 4,100 e.h.p. corresponding to 99.6 h.p. per litre, the specific weight being reduced to .93 lb per e.h.p.

Two further factors which arise from the nature of the operating cycle are worthy of mention.

Firstly, the engine is able to run effectively on a wide range of fuels including diesel fuel, for which it was originally designed, kerosine and wide-cut gasoline.

Secondly, the engine is practically insensitive to increases in ambient temperature, losing less than 2% of its take-off power for each 10° C increase in temperature.

Hydro-Electric Development in Portugal.

PART TWO

Part One of this article was published in the June 1954 issue of this Journal, volume thirteen, number six, page 223.

RIVER CAVADO POWER SCHEME

As already stated, the other scheme for which a contract was placed in 1946 with the 'British Group' for plant and equipment was on the River Cavado in the northern part of the country. This river flows into the sea about 30 miles north of Oporto, and some 40 miles from its mouth it is joined by a tributary, the River Rabagao, across which a dam has been built; the reservoir so formed is known as Venda Nova. Water from the reservoir flows through a tunnel and pipeline to the power station at Vila Nova, where it is discharged into the River Cavado. The power station has been equipped with three generating units having a total capacity of 96 MVA, and provision has been made for the installation of a fourth unit when another dam has been built on the River Cavado, upstream of Vila Nova, at Paradela. It is of interest to note here that this extension was approved by the Portuguese Government in 1952 in the same development plan that included the Bouca project on the River Zezere. The installed capacity at Vila Nova will then be 156 MVA.

The chief characteristics of the River Cavado development are given in Table II.

The Salamonde project is now operating, Canicada is under construction, and Paradela is included in the new construction plan.

From the plan and profile in Fig. 16 it will be seen that the dam at Venda Nova on the River Rabagao is about 4 km (2½ miles) from the power station at Vila Nova, the turbines of which discharge into the River Cavado. The dam is 97 m (318 ft) high and the water which it impounds is conveyed to the power station by a tunnel and steel pipeline. When the dam at Paradela (near the confluence of the rivers Pitoes and Cavado) is completed, a further tunnel will be driven connecting the reservoir so formed with Vila Nova, thus enabling the installed capacity of this station to be increased by 60 MVA. At present Vila Nova is equipped with

Table II

RIVER CAVADO POWER SCHEME

Project	Height of dam		Length of pipe-line and tunnel		Capacity of reservoir		Maximum gross head		Installed capacity	Firm annual output
	Metres	Feet	Km	Miles	$m^3 \times 10^6$	Acre-feet	Metres	Feet	MVA	$kWh \times 10^6$
Venda Nova	97	318	3·5	2·06	92	74,700	415	1,360	3×32	200
Salamonde	75	246	2·1	1·31	55	44,700	127	417	2×25	200
Canicada	76	249	7·8	4·85	138	112,000	121	397	2×32	260
Paradela	105	345	10·6	6·6	140	113,000	449	1,475	1×60	260

three units having a total capacity of 96 MVA. At the Venda Nova dam a diffusing disperser valve is installed, similar in design to those on the scour culverts at Castelo do Bode. It is 2·6 m (8 ft 6 in) in diameter, designed for hand and electrical operation, and is capable of discharging 120 cubic metres (4,230 cubic feet) of water per second under a 100 m (328 ft) head.

At the Vila Nova end of the tunnel there is a portal valve of the self-closing 'Straightflow' design, 2·13 m (7 ft) bore. This valve is operated by the action of water pressure and is generally similar to the corresponding valves described in connection with Castelo do Bode. It is provided with a paddle trip for emergency closing, consisting of a gunmetal rod suspended in the waterway. From this valve the pipeline, which is about 854 m (2,800 ft) long, descends the hill to the distributing piping and thence to the turbines.

The pipeline is in three main sections, 2·40 m (7 ft 10½ in), 2·26 m (7 ft 5 in) and 2·1 m (6 ft 11 in) outside diameter, with a thickness varying from 19 mm (¾ in) to 51 mm (2 in). It is made of mild steel and the pipes and bends were welded together on site and afterwards tested at a pressure equal to 175 per cent. of the static head. The distributing piping passes along the side of the power station with branches to the turbine inlet valves. Within

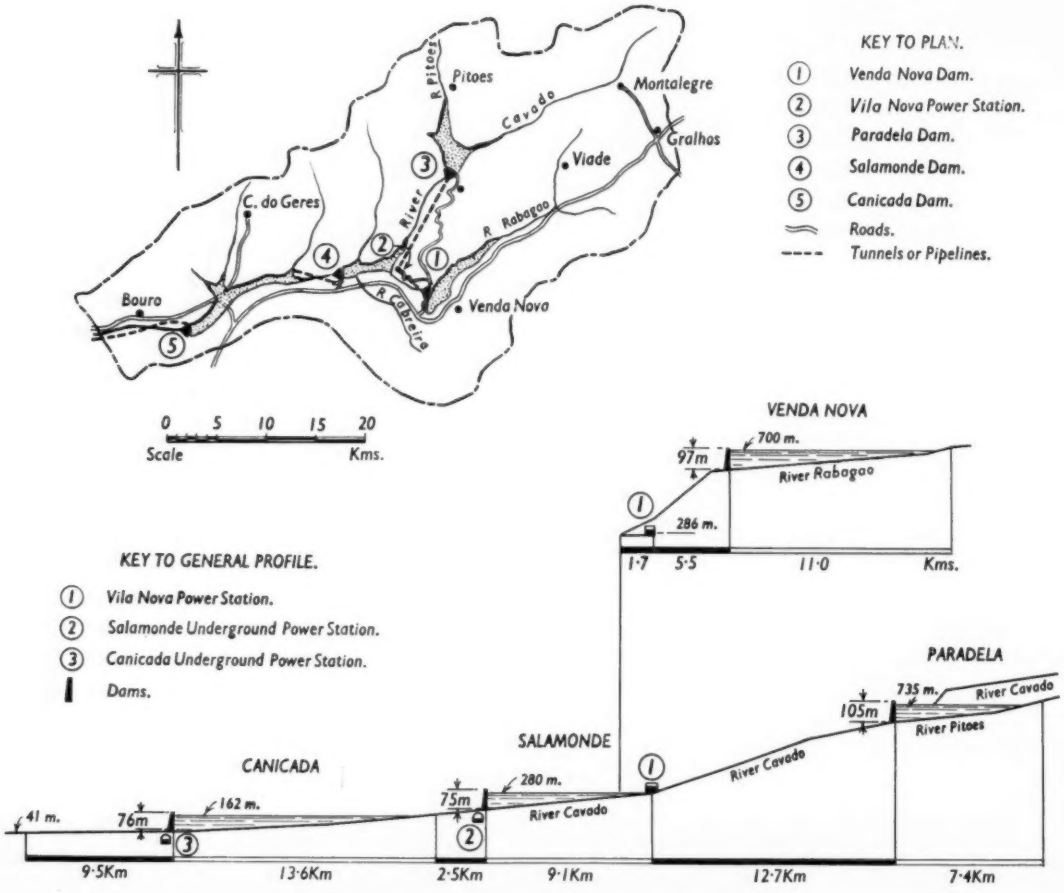

Fig. 16.—Plan and profile of River Cavado development

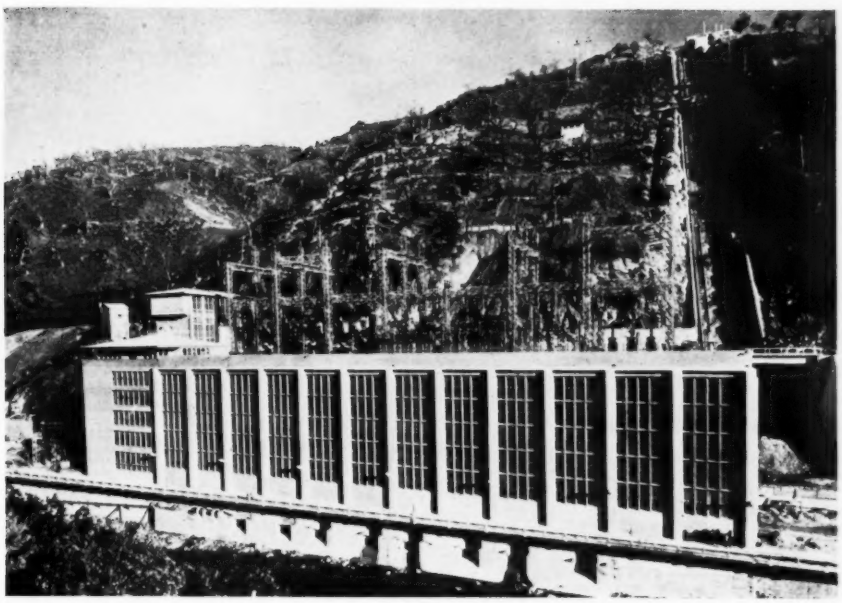

Fig. 17.—Vila Nova power station

this piping there is an isolating 'Straightflow' valve 1·2 m (47¼ in) bore, combining compactness with smooth bore flow which is important because of the high velocity at this point. There are six turbine inlet valves, two for each of the three twin turbines, and they are also of the 'Straightflow' type. They are 825 mm (32½ in) bore, and because of the high head have been made of cast steel. Fig. 17 is a view of the station showing the pipeline and the further track which is available for another pipeline when the Paradela extension comes into operation.

VILA NOVA POWER STATION

A plan of part of the station arrangement is given in Fig. 18, and Fig. 19 is a view of the machine room. The main generating units are installed down the centre. On the downstream wall, opposite each unit, are motor-driven exciter sets and their control boards. On the upstream side are the valve control cubicles, the governor oil pressure pumping sets, the bearing oil pumping sets and the gauge panels. A turbine-driven auxiliary generating set is placed at one end of the station and space is left for a further auxiliary set at a future date. The main valves are below floor level, covered by chequer plating which can be removed for crane access.

At one end of the station is the control room, which overlooks the machine hall, and on the upstream side are the 11 kV and 33 kV switchgear rooms and the cable galleries. The 165 kV switchgear is on a level with the roof of the power station, slightly upstream and against the steep slope of the hillside (Fig. 17).

The sets are started up and shut down manually, but full control is centralised in the control room. Synchronising takes place automatically when voltage and synchronism have been correctly adjusted.

Turbines

Because of the higher head, the three turbines at Vila Nova (Figs. 19 and 20) are very different from those at Castelo do Bode. They are horizontal shaft, four-jet impulse turbines with twin runners, one overhung at each end of the generator shaft and each actuated by two jets. Under a gross head of 395 m (1,300 ft) each turbine is rated at 36,000 h.p., and under a maximum gross head of 415 m (1,360 ft) each turbine will give 39,000 h.p. The normal speed is 428 r.p.m. and the machines are

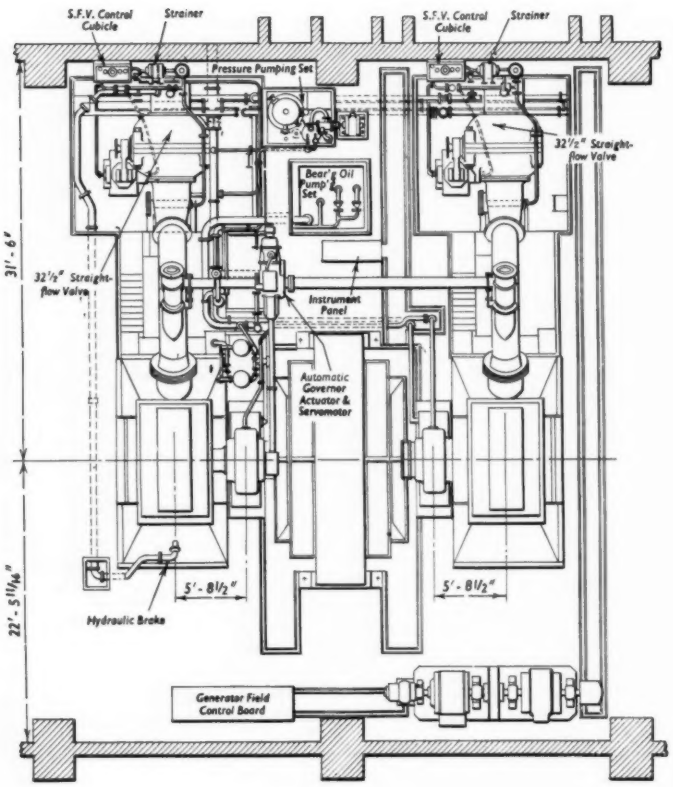

Fig. 18.—*Plan showing arrangement of part of Vila Nova power station*

designed to withstand a runaway speed of 792 r.p.m. under the maximum head.

The runners are of stainless steel cast in one piece, that is, with integral buckets. The casings are fabricated and each turbine has two inlet bends, each of which is equipped with a nozzle, spear and the jet disperser of the ' Seewer ' governing device. Within the spearhead there are a number of slanting streamline guide vanes, normally withdrawn into the head and controlled by the governor by means of a distributing valve and an oil pressure servo-motor. When a sudden drop in load occurs, the vanes are projected into the water stream and diffuse the jet, thus destroying the energy imparted to the buckets of the runner. This device ensures sensitive high-speed governing because little energy is required for introducing it into the jet, which is diffused very rapidly. In Fig. 21 is reproduced a photograph of a model of one of the turbines with its inlet valve ; it will be noted that the diffusing vanes are shown projecting from the spearhead.

A system of hydraulic braking enables the set to be brought rapidly to rest : water under pressure is tapped from upstream of the inlet valve and made to impinge on the backs of the buckets through a nozzle mouthpiece and control valve. This device can also be seen in Fig. 21.

Governing is by automatic oil pressure equipment with an actuator similar to that at Castelo do Bode. In this case, however, the governor pendulum has a mechanical drive consisting of a spur and bevel gear connecting the turbine shaft with the governor head by a shaft into which a flexible stainless steel plate has been inserted. This plate provides a combination of rigidity and flexibility and thus eliminates vibration and the detrimental effect of temperature changes. The gear drive operates entirely in an oil bath.

Safety gear is provided to shut down the turbine by the operation of the diffuser in the event of governor drive failure, underspeed or oil pressure falling below a certain figure. As already mentioned, there are six valves for the three twin turbines. Each valve has one servo-motor, and operation is by water pressure. Despite the high-pressure due to the head, both the service seals and the emergency isolating seal incorporate rubber as the staunching medium, which is adjustable to give complete drop-tightness.

The turbine of the auxiliary generating set is a horizontal-shaft impulse machine, overhung at one end of the generator shaft and actuated by one jet. It is rated at 1,150 h.p. under 365 m (1,200 ft) gross head and 1,300 h.p. under 415 m (1,360 ft) gross head and runs at 1,000 r.p.m. It has a cast steel

runner with integral buckets, and an automatic oil pressure governing system.

Generators

The generators are horizontal machines having a rated output of 32 MVA (25·6 MW) at 0·8 lagging power factor, 11 kV, three-phase, 50 c/s. They are connected through step-up transformers to the 165 kV transmission system and are capable of an output of 20 MVA at zero leading power factor for line charging. The main and pilot exciters are separately driven by an induction motor supplied through unit transformers from the generator terminals. During the running-up period the driving motor supply is obtained from the station common services board, which in turn, is energised from the 875 kVA, 0·8 power factor, 1,000 r.p.m. station auxiliary set or, in an emergency, from a source external to the station via auxiliary transformers provided for this purpose.

The main generators are of conventional design with two pedestal bearings and an enclosed ventilation system with water-cooled air coolers mounted in the generator pit and constructed in four units to reduce withdrawal space. Although the normal running speed is 428 r.p.m., the generators are designed to withstand the stresses produced at the turbine runaway speed of 792 r.p.m.

The exciter set consists of main and pilot exciters and a flywheel driven by a 170 h.p. induction motor at 1,494 r.p.m. The flywheel is provided to smooth out the field-forcing peaks and to provide the stored energy for maintaining the excitation should the generator voltage and hence the driving motor supply voltage collapse under fault conditions.

The common services board is normally supplied from an impulse-turbine-driven auxiliary set running at a normal speed of 1,000 r.p.m. and designed to withstand the stresses that may occur at the turbine runaway speed of 1,900 r.p.m. To provide the inertia required for satisfactory governing, the inherent flywheel effect of the generator is augmented by a flywheel mounted next to the generator between the two bearings.

Fig. 19.—*Machine room of Vila Nova power station*

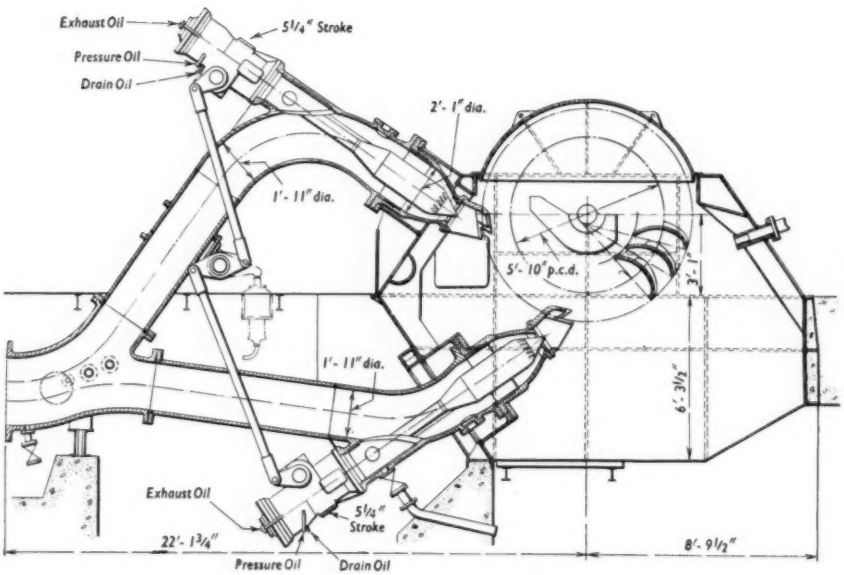

Fig. 20.—Sectional elevation of 39,000 h.p. turbine

Generator Transformers

A direct-connected group of main transformers is installed for each of the three generators. Transport between the port of entry in Portugal and the site was difficult and, to meet the limitations, three-phase groups of single-phase transformers had to be adopted.

Each 30 MVA three-phase group comprises three 10 MVA single-phase transformers connected delta on the lower voltage windings and star on the higher voltage windings for a voltage ratio at full load, 0·85 p.f., of 11/165 kV. The neutral point of the 165 kV windings is solidly earthed. Plus and minus 5 per cent. tappings are provided on the high-voltage windings, and the tappings are connected to an externally operated off-circuit tapping switch.

The transformer losses are dissipated with forced oil circulation through water-cooled coolers. An oil pump and cooler are installed for each three-phase group, and 100 per cent. stand-by pump and cooler capacity is provided. Transfer is made by valve adjustment. The cooler circuits are complete with the usual oil and water flow alarms. The total weight of each single-phase transformer is 35 tons ; stripped for transport the weight is 23·5 tons.

Switchgear

11 *kV Connections.*—Each of the three 32 MVA generators is connected solidly to its main 11/165 kV step-up transformer, with a tee-off to its 11,000/380 V unit transformer. These connections take the form of bare copper conductors having a rating of 2,000 amp, mounted on post insulators with phase barriers, running through horizontal and vertical ducting to the 11 kV switchgear on the fourth floor of the power house. To give greater flexibility for emergency and maintenance purposes, transfer busbars run on the wall above the switchgear for the whole length of this floor, and selector links enable any generator to be connected to any main transformer bank or to be loaded on to a water resistance test tank. The control and protective circuits are provided with similar links to ensure correct operation under all conditions. Each switchgear equipment associated with a generator contains the above-mentioned selector links for the main circuit, isolator links for the unit transformer, and instrument transformers for protective circuits and metering duty ; these, together with the unit transformers themselves, are enclosed by expanded metal screenwork on three sides, with the wall at the rear.

The main 11 kV circuits are connected through

roof bushings to the low voltage sides of the respective transformer groups and, where delta connections are made, consist of bare copper supported by outdoor post insulators on steel pipe framework.

165 *kV Outdoor Substation.*—Situated on a level with the fifth floor of the building is the outdoor substation (Fig. 22), consisting initially of three generator circuits, bus-coupler and outgoing feeder to Oporto, with space for one future generator and one future feeder (Fig. 23) and all arranged in line abreast. Construction is of high level design with the duplicate busbars arranged side by side at a height of 63 ft and comprising stranded copper conductors suspended between strain insulators. Busbar selector isolators, arranged for off-load operation with electrical interlocks, are mounted below the busbars and connected back-to-back by copper-clad steel tubes supported by post insulators. Feeder isolators are provided with mechanically interlocked earthing contacts, and all isolators are local-mechanically controlled and electrically interlocked with the respective circuit-breakers. The supporting structures are of galvanised steel with a maximum height of 80 ft, the outgoing feeder transmission line to Oporto spanning the Cavado River.

165 *kV Circuit-Breakers.*—These circuit-breakers (Fig. 24) are air-blast equipments, each rated at 600 A and having a breaking capacity of 2,500 MVA at 165 kV. Three interrupter units are used in series per phase, mounted to form a single vertical column immediately above the housing containing the operating cylinders for the sequence making switch which operates in the vertical plane. Each interrupter unit is shunted by a non-linear resistance for suppression of switching overvoltages and to ensure effective voltage division. The speed of operation from trip impulse to arc extinction is of the order of three cycles.

Current transformers are included as an integral part of the circuit-breakers ; they are oil-immersed, cross-coil, porcelain-clad wound units, with multiple

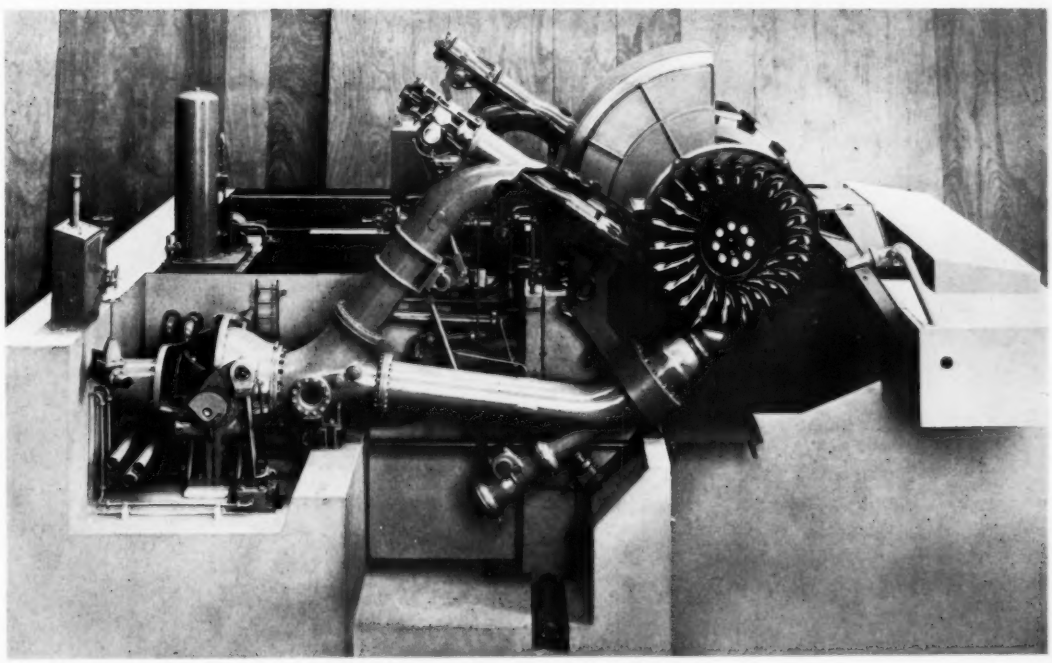

Fig. 21.—*Model of* 39,000 *h.p. turbine*

cores and secondaries as required for the various duties. The Oporto feeder breaker is equipped for single-phase and three-phase, high-speed, single-shot auto-reclosure, with 'dead time' adjustable from 20 to 60 and 10 to 20 cycles respectively. Relay equipment and selector control are mounted in the main control room.

Compressor Equipment.—The plant for the operation of the circuit-breakers is housed on the floor immediately below the outdoor substation and is of adequate capacity for the planned extensions. Two main storage vessels are provided, each fitted with duplicate compressors automatically operated by rise and fall of pressure. The vessels are inter-connected and all pipework duplicated for ease of repair and maintenance. These vessels store air at 600 lb per square inch, which is expanded to 300 lb per square inch through reducing valves to the bus piping systems, thereby obtaining the necessary dryness before operating the switchgear. Feeds from the system are also taken to the 30 kV switchgear and turbine governors through reducing valves at a pressure of 250 lb per square inch.

30 *kV Switchgear.*—For bringing in a separate 30 kV supply from Ermal, and also for supplying power to the local settlement and valve houses at the dams, the scheme includes two 33 kV, 400 A air-blast circuit-breakers similar in principle to the 165 kV breakers but having only one interrupter unit per phase and a breaking capacity of 350 MVA at 33 kV. These circuit-breakers, together with associated isolator gear, insulators, current transformers, etc., are mounted in stonework cells

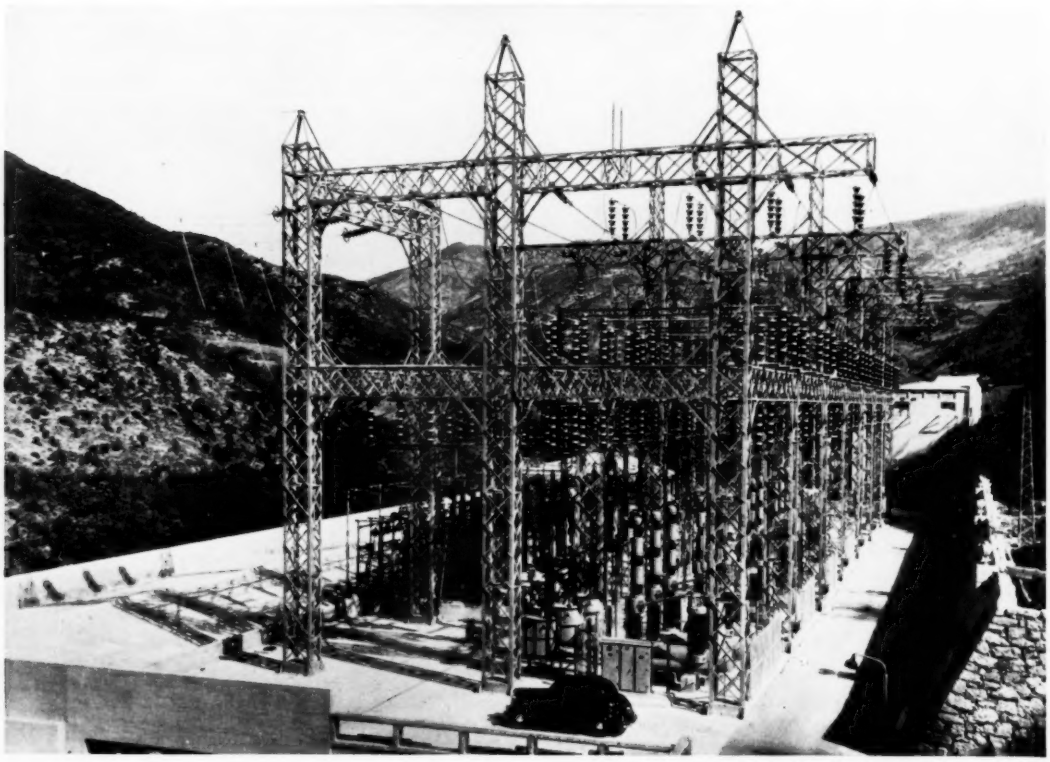

Fig. 22.—Vila Nova 165 kV outdoor substation

having expanded metal screenwork and doors suitably interlocked. Two feeds are also taken to the auxiliary transformers for interconnection with the common services board. Remote control, indications and relay equipment are in the control room.

Station and Unit Auxiliaries.—Generation is at 380 V, three-phase, three-wire, independent of any external source.

The common services board is of single busbar construction and comprises remotely operated 25 MVA air circuit-breakers, rated between 600-1,600 A according to duty, for auxiliary generators, bus-section and auxiliary transformer circuits. Provision is made for two auxiliary generator circuits, although initially one impulse-turbine-driven 875 kVA auxiliary generator only is connected, the bus-section switch being closed. Extensions to the left and right are of high rupturing capacity, ironclad combination fuse-switch-gear, for station supplies which are as far as possible duplicated. Lighting supplies are taken from this board to three-wire/four-wire transformers to reduce fault capacity and also to avoid having a four-wire system throughout.

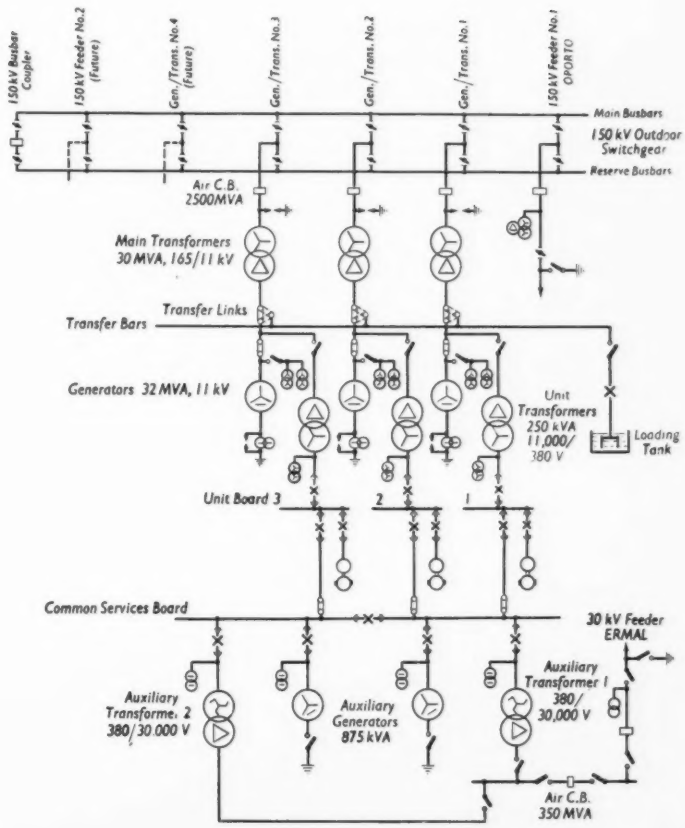

Fig. 23.—Key diagram of Vila Nova power station

The unit boards are of similar composition to the foregoing and provide power for duplicate generator lubricating oil pumps, transformer oil pumps and the separately driven exciter set associated with each main generator. The circuit-breakers take the feed from the unit transformer and they provide an interconnection with the common services board.

Control Room.—The main control board (Fig. 25) consists of vertical sheet steel cubicles arranged on three sides of a rectangle with the 165 kV section and unit auxiliary control in the centre, the 30 kV and 380 V sections being mounted on either side. Each control panel has its circuit mimic diagram with illuminated semaphore indicators and illuminated circuit labels. The centre panel of the 165 kV section comprises a central alarms annunciator with telephonic relays, above which are mounted the ' comparator ' dials of the master frequency equipment. There are two swinging synchronising panels, one being used for 165 kV switchgear only and the other for the 30 kV and 380 V switchgear, check synchronising facilities being included.

A seven-panel desk control board of sheet steel construction is positioned centrally and includes control of the water resistance test tank and future main and auxiliary sets. The relay, metering and

automatic voltage regulator cubicles are of similar construction and are suitably placed in the control room with respect to the various boards.

Another separately mounted hydraulic control cubicle gives alarm and indication of the Venda Nova and future Paradela dams, pipelines and turbine inlets, on a mimic diagram with illuminated semaphore indicators. It is also possible to close any valve in an emergency from this panel, the control scheme being supplied at 24 V D.C. Dam level and pipeline water flow indicators complete the equipment.

Protection.—Briefly, the scheme of protection is as follows :—

(*a*) The generator neutrals are earthed through a voltage transformer across which is connected a surge arrester. The secondary side of the voltage transformer is connected to a relay insensitive to predominant harmonics, the operation of which causes the set to shut down.

(*b*) The main generator is provided with instantaneous circulating current protection, with inverse back-up overcurrent from current transformers in the star connections, and over-voltage protection. A rotor earth fault alarm is provided.

(*c*) The main transformer protection is by the biased differential method with a slight inverse characteristic to ensure immunity from operation by magnetising in-rush at switching. Buchholz relays and oil and winding temperature indicators and alarms are fitted. The transformer high-voltage neutrals are solidly earthed and include a current transformer for operation of an induction relay with inverse characteristics. The 165 kV station surge arresters are connected as near as possible to the outgoing terminals.

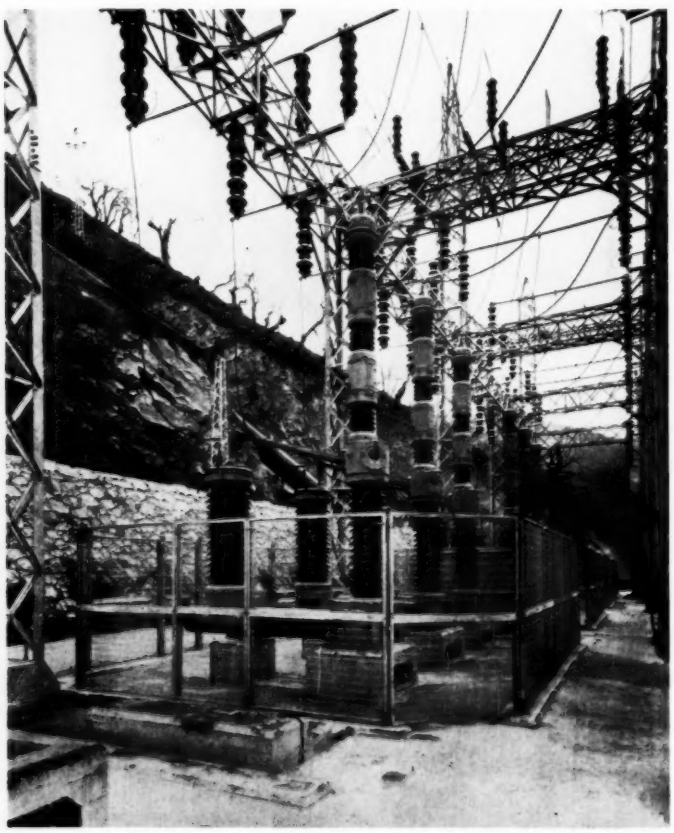

Fig. 24.—The 165 kV air-blast circuit-breakers at Vila Nova

(*d*) The 165 kV feeder to Oporto is fitted with an induction back-up over-current relay only, by the 'British Group.' Impedance relays which are suitable for single- and three-phase auto-reclose were the subject of a separate contract ; these protective relays, however, were mounted and wired on panels provided by the 'British Group' as part of the main switchgear contract.

(*e*) The unit transformers are fitted with high-voltage inverse over-current and low-voltage balanced earth fault protection.

(*f*) The auxiliary generators have instantaneous circulating current relays with inverse back-up over-current protection.

Ancillary Equipment.—Ancillary equipment includes a 230 V station battery equipment for main control circuits and a 24 V battery for semaphores, hydraulic control panel, master frequency equip-

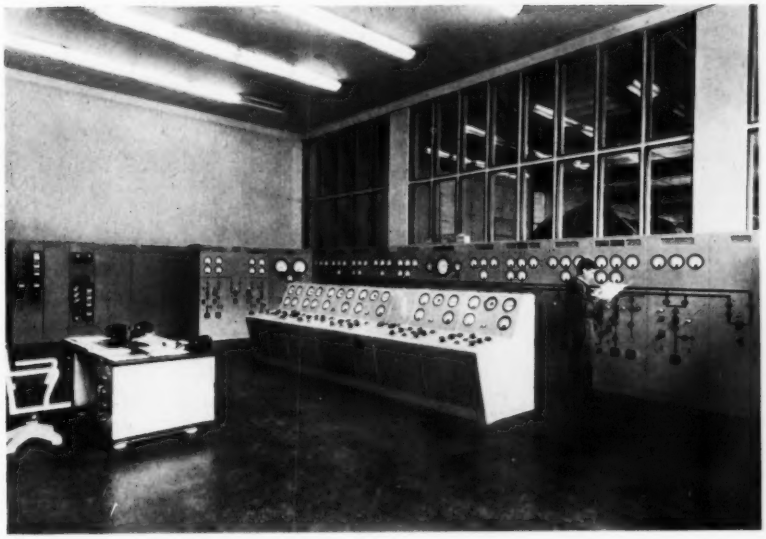

Fig. 25.—The control room

ment and automatic telephone exchange, all supplied by the ' British Group.' A flat-back battery distribution board is supplied, on which is mounted a change-over contactor to give 230 V D.C. emergency illumination supply on the occurrence of mains failure.

General Station Equipment

Cranes.—The power station crane is designed to handle a maximum working load of 110 tons and has the motions of main and auxiliary hoisting (25 tons), traversing and travelling, operated by four electric motors. Automatic electric brakes and upright reversing drum controllers are provided.

In the transformer workshop there is a 35-ton travelling crane with motions of main and auxiliary hoisting (5 tons), travelling and traversing, operated by four electric motors. In the valve house there is a 40-ton crane arranged for hoisting at two speeds, travelling and traversing ; it is operated by three motors.

Aerofoil Flow Recorders.—An aerofoil water flow recorder is installed in the pipeline for flow indicating and recording. The aerofoil assumes a different angle of incidence for varying rates of flow and is constrained by a simple coil spring. An extension of the pivoting shaft operates the indicating and recording apparatus.

Sundries.—There are also a dam water level indicator, cooling water pumps and piping for generator and transformer cooling water, purifying and testing equipment for turbine, generator and transformer oil, and sprinkler fire-fighting equipment.

Principal Manufacturers

The principal manufacturers of the plant and equipment for the Vila Nova power station contract were as follows :—

Main and auxiliary water turbines and turbine inlet valves, The English Electric Company, Ltd. ; main and auxiliary generators and main, auxiliary and lighting transformers, Metropolitan-Vickers Electrical Company, Ltd. ; power station and substation switchgear and control gear, disperser valve, portal valve and isolating valve, waterflow measuring equipment, The English Electric Company, Ltd. ; pipelines and distributing piping, the South Durham Steel and Iron Company, Ltd. ; power station, workshop and valve house cranes, Sir William Arrol and Co., Ltd. ; oil purifying equipment, Alfa Laval Company, and Streamline Filters, Ltd. ; ' Mulsifyre ' fire-fighting equipment, Mather and Platt, Ltd.

Some Notes on Steam Turbine Development

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This article is based on the paper read by the author before the Rugby Engineering Society on the 18th of November, 1953.

Great developments in steam turbine construction have taken place in the last ten years. Scarcity of power generating plant and of sites for power stations, increasing cost of fuel, and progress in the art of engineering have been responsible for this. The 30,000 kW machines for 600 p.s.i.g. and 850°F at the turbine stop valve, which before the last war were considered the most economic 'large machines' for central power stations, are now half way down the list of standard sets headed by 100,000 kW units for 1,500 p.s.i.g. and 1,050°F at the stop valve.

It is a measure of progress in design that such very advanced conditions have been incorporated in Recommended Standards tables. A pressure of 1,500 p.s.i.g. is one that would be measured at the base of a column of water approximately as high as Snowdon, and steel glows dark red at a temperature of 1,050°F.

Steam turbine manufacture in The English Electric Company's Rugby works began 51 years ago, and in that time steam conditions and sizes of units have greatly changed. Modern average coal consumptions compared with those in the days of the first turbines in 1903 have been reduced from 5.6 lb to about 1.4 lb per kW, and these gains are due to improvements in the cycle efficiency, prime mover efficiency, loading and operation efficiency, boiler efficiency and in auxiliary and pumping power. The changes in cycle efficiency may be shown as being due, in the following approximate proportions, to:

(*a*) increase in temperatures from 600°F to 1,050°F	13% - 17%
(*b*) increase in pressures from 200 p.s.i.g. to 1,500 p.s.i.g. ..	12% - 14%
(*c*) feedwater heating	6% - 12%
(*d*) reheat cycle (where applicable)	2% - 6%

In 1926 the systems of individual supply undertakings in Britain were formed into an interconnected 'grid' under the Central Electricity Board, but a closer integration was made possible by the establishment of the British Electricity Authority in 1948. The Authority can review the power requirements of the country as a whole and arrange the disposition of suitable power station sites. Land, water and coal are required for a steam power station, and the first two together, for this purpose, are becoming difficult to find in Britain. This will be easily appreciated when it is realized that a third of a mile square may be required for a medium-size power station on a river site where the cooling water necessary would be of the order of 12 million gallons per hour. For this reason it is advisable to place the most economical stations at the favourable sites (those becoming difficult to find) and operate them as continuously as possible. It is advisable to use the less favourable sites for building stations for less efficient but fairly continuous service. Least suitable sites should be used for distinctly intermittent (or peak) service. These three kinds of stations are represented in the British Standards table of recommended sizes by:—

- 100 MW turbines, 1,500 p.s.i.g. and 1,050°F, for most efficient service;
- 60 MW turbines, 900 p.s.i.g. and 900°F, for less efficient service;
- 30 MW turbines, 600 p.s.i.g. and 850°F, for intermittent service.

The size of the station for an individual supply centre depends of course on the maximum load to be carried. In the case of an interconnected system the problems are much more complicated; calculations may be made summing up the relevant costs of generation and costs of interconnection, and

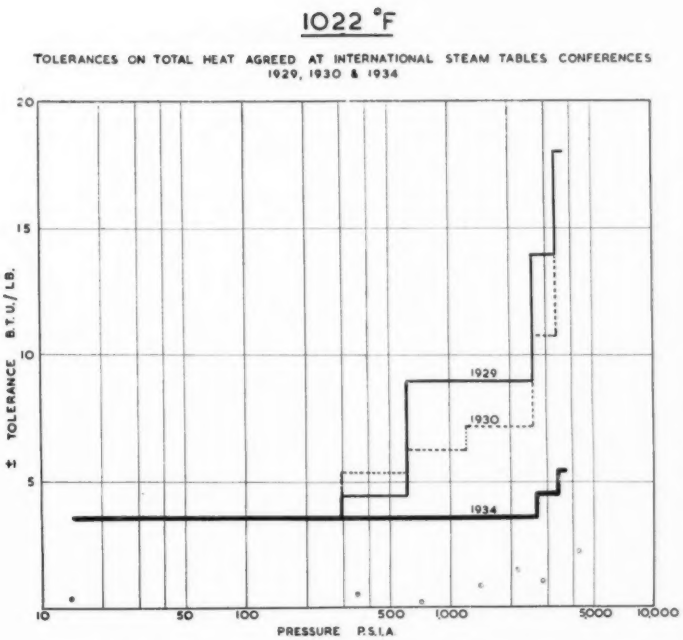


Fig. 1. *Tolerances on total heat,* 1929 *to* 1934

finding the optimum by differentiating with respect to plant size. Following Ricard's paper[1] to the World Power Conference in 1950, Donkin and Margen[2] made such calculations for the British Grid and reached the conclusion that with increasing confidence in the large unit which would come from operational experience, and with an increase in prices generally, 100 MW sets would become the most economical single units in Britain.

On the whole it may be said that the type and spacing of steam turbine stations and the size of individual units installed therein would depend on the:—

(1) size and character of the network to be served as well as the sites available;
(2) reduction in cost/kW of plant, and reduction in generating charges associated with increasing sizes of units; this covers such factors as the cost of coal, rate on capital, load factor, cost of high temperature and high pressure construction, etc.;
(3) degree of electrical interconnection and standby capacity;
(4) necessity to conserve coal supplies.

This is the background against which the development of steam turbine designs proceeds in this country, and it is very much the same in the United States and in Europe. There, too, a great deal of interconnection of individual smaller undertakings has taken place, and the general economic conditions have encouraged the trend towards very large turbine units operating at high temperatures. In the U.S.A. recently, orders have been placed for 250 MW single units to operate at 1,050°F at the turbine stop valve, and 200 MW units are under consideration in Britain.

Some of the problems associated with the development of modern steam turbines are briefly discussed in the following sections of this article. The first three sections deal with some thermal and mechanical limitations to design, presenting problems that need to be solved to achieve safe operation and efficiency at a cost consistent with these levels of safety and efficiency. The fourth section deals with a new approximate method of calculating disc stresses. In conclusion, some current designs are discussed and new projects mentioned.

I.—STEAM PROPERTIES

Steam turbine developments towards better cycle efficiency, previously mentioned, had one rather astonishing consequence. They brought us to the very limit of data available for steam flow design, indeed, the standard 1,500 p.s.i.g., 1,050°F machines are outside the range covered by charts and tables commercially produced in Britain. The problem of extending the tables of steam properties is being tackled in this country under the auspices of the British Electrical and Allied Industries Research Association (B.E.I.R.A.) and it may be of interest to give a brief summary of this particular aspect of steam turbine research.

[1] *Ricard, J. 'Concentrations des puissances dans la production d'energie thermique.' Fourth World Power Conference, Section E1, Paper 3.*

[2] *Donkin, B., and Margen, P. H. 'Economic plant sizes and boiler groupings on the British grid.' 'Engineering,' 1 and 8 Feb., 1952.*

Until 1929 steam properties were investigated in several countries of the world without any attempt being made officially to correlate the available data. In July of that year the B.E.I.R.A. called a meeting in London inviting engineers and physicists who were connected with or engaged in measurement of steam properties. Countries represented on that occasion were Great Britain, U.S.A., Germany and Czechoslovakia. It was decided there to collect steam data from the four participating countries and arrange a steam table where at each point a mean value of figures from each country was recorded together with a tolerance covering the other figures. With the variety of experimental methods in use by the different investigators a large tolerance in the high temperature, high pressure regions was only to be expected. It was appreciated also that in tabulating pressures, temperatures, specific volumes and total heats, the method of recording a mean value of four proposals for each of these data would not be entirely correct. Theoretically, pressure and temperature alone would be sufficient to define other properties of steam at that point, and it would therefore be possible that for a given pressure and temperature incompatible volumes and total heats would be recorded. The choice of the tolerance had to be such as to cover this particular difficulty.

A sub-committee was elected from the meeting, and under the chairmanship of Mr. I. V. Robinson[3] prepared the 'skeleton tables' for which generally acceptable units of measurement were chosen. Metric units were adopted as being in general use by physicists of all countries; however, the heat unit chosen had to be free from gravity effects and changes in the specific heat of water. An international kilo-calorie was created which was equivalent to 1/860 of a kilowatt-hour, and differed from the kilo-calories accepted in different lands by less than $\pm 0.05\%$. Conversion factors to the F.P.S. system were also generally agreed. It may be of interest to add that in 1950 international agreement (subscribed to by the Royal Society and B.E.I.R.A) had been reached on the use of the kilojoule as the common heat unit in place of the British thermal unit and the kilo-calorie. 1 kilojoule per second is equal to 1 kW, and the rate of conversion is 1 B.Th.U. = 1.055 kJ.

[3] *Then Turbine Group Secretary of the British Electrical and Allied Manufacturers' Association.*

It is intended that as time proceeds, this alteration in units should be accepted by engineers, and in the interim various scientific bodies should be printing their proceedings with B.Th.U. and kJ units side by side.

Fig. 1 shows the tolerances on total heats which it was found necessary to accept in 1929, over a range of pressures at 1,022°F. Stop valve pressures and temperatures of the order of 400 p.s.i.g. and 750°F were fairly common at that time, and an error of $2\frac{3}{4}$ B.Th.U/lb could be associated with a heat drop of, say, 400 B.Th.U/lb or nearly 0.7%.

The sub-committee pressed on with their programme which led to the second International Steam Tables Conference in Berlin in June 1930. This was sponsored by the Verein Deutscher Ingenieure and included Great Britain, U.S.A., Germany, Switzerland, Czechoslovakia and Sweden. During this conference more tolerances were narrowed and the skeleton tables were enlarged by several sets of intermediate figures.

The third and last International Steam Tables Conference took place in New York in September 1934, sponsored by the American Society of Mechanical Engineers. A good deal of new experimental work had been done by then to check and enlarge the existing skeleton tables and particularly to reduce the large tolerances accepted before. Over 400°F, saturation figures were enclosed within tolerances equal to one-half to one-tenth of the previous tolerances; over 1,400 p.s.i.g., specific volumes were agreed to within $\pm 0.2\%$ instead of the Berlin figures of 1% to 3%. Finally, total heats of superheated steam were accepted within half of the tolerance range of the previously accepted figures. New work was planned and the organization of a fourth conference was left with the Masaryk Academy in Prague. This has never been called.

The tolerances at 1,022°F agreed in New York are shown in Fig. 1, and Fig. 2 shows these and other tolerances related to a 1939 Callendar chart.

The present position regarding steam tables is as follows:—

Country	Tables Used	Extent of Tables p.s.i.a.	Extent of Tables °F
Great Britain..	Callendar	3,200	1,000
U.S.A. ..	Keenan and Keyes	5,500	1,600

U.S.S.R.	..	Vukalovitch	4,266	1,292
Germany	..	Koch	4,266	1,022
Japan ..	..	Japanese Society of Mechanical Engineers	4,266	1,112

In the range of 4,260 p.s.i.a. and 1,292°F the divergences between the various tables amount to more than 1%, and the need for further measurements therefore becomes clear. Such work is being carried out at the Imperial College of Science and Technology in London under the auspices of the B.E.I.R.A., and Professor Newitt is in charge of research intended to enlarge the tables to 6,000 p.s.i.a. and 1,400°F.

This section should not be closed without some indication of the practical significance of these chart discrepancies. Although steam tables and charts are not 100% accurate and cannot be prepared with the same degree of accuracy as, say,

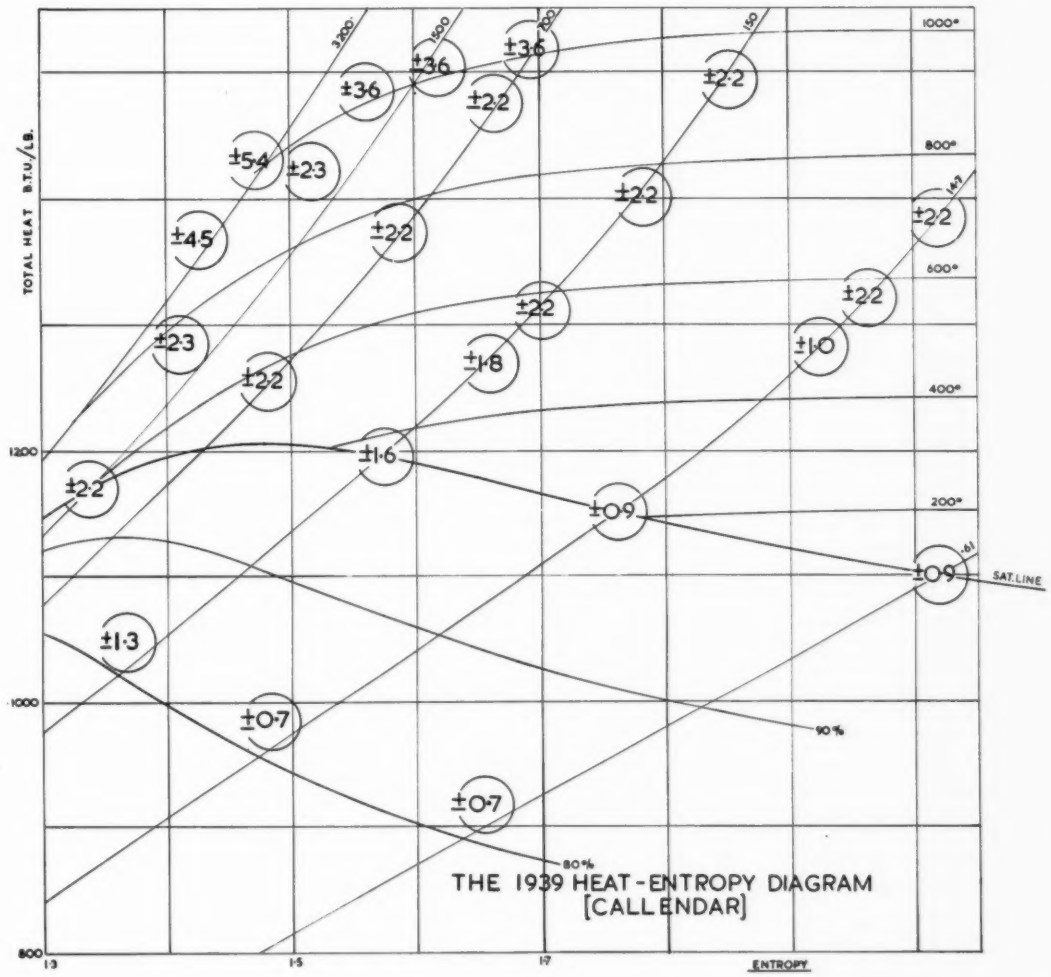


Fig. 2. Approximate tolerances on total heat agreed at the International Steam Tables Conference in 1934 (± *B.Th.U/lb*)

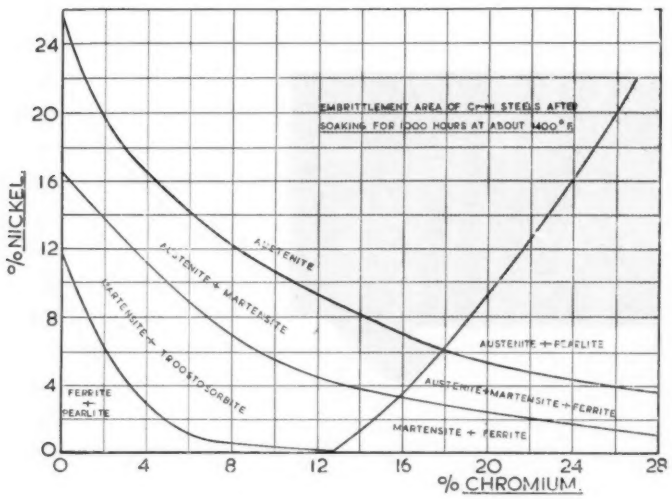


Fig. 3. Hoch's diagram

logarithmic tables, they are, of course, in most instances perfectly satisfactory for commercial steam turbine work.

Chart and table discrepancies tend to affect design in a variety of ways, but particularly the calculation of heat consumption which involves total heats of working fluid entering and leaving the boiler and the total heat of steam at the end of expansion through the turbine.

It is estimated that within the range of turbines built today, 0.1% difference in heat consumption could easily be calculated by using one country's tables instead of another's. With large steam turbines the value of each 1% of efficiency is considerable (£100,000 to £200,000) and this particular 'tables effect' may therefore be of appreciable interest in competitive bidding.

Since tolerances required in the high pressure high temperature range are much greater than in the low pressure low temperature range, the discrepancies in steam tables would influence the so-called topping turbine projects much more than condensing turbine projects, and these chart inaccuracies could be responsible for dissipating a high proportion of the gain expected from such developments.

II.—STEEL AT HIGH TEMPERATURE

Progress in initial steam conditions towards higher temperatures and pressures is controlled by several factors already mentioned. One definitely limiting factor is a general deterioration of properties of steel at high temperatures. Steel changes from a proverbially reliable metal at room temperature to a less reliable metal at high temperatures due to alteration in microstructure. Thus with mild steels, high temperature causes spheroidization of the flake carbide in the pearlite which reflects on the mechanical strength of the metal. This effect becomes pronounced above 800°F and is retarded by additions of molybdenum.

As is the case with most advanced developments, there is a certain amount of controversy about the problems of high temperature materials, and although an attempt is made to give a balanced picture of the situation, many remarks in this section have to be viewed with this reservation in mind.

Early turbine rotors were built up of several steel parts which were joined by shrinkage, large buttress threads or by bolting. As temperatures and speeds began to rise there was a desire to avoid using built up rotors, which focused the attention of engineers on the production of large forgings. The early 'twenties saw the advent of an all-solid steel rotor in which the required discs were gashed out of a solid billet of steel. Up to about 750° - 800°F plain carbon-steel rotors have been perfectly satisfactory, but above that temperature creep became a very distinct limitation to safe design. This continuous stretching of steel under stress at temperatures up to 900°F is satisfactorily resisted by small additions of molybdenum; chromium and nickel are added primarily to improve properties at room and medium temperatures. Beyond 900°F other alloying additions like vanadium must be used to stabilise the steel. Very extensive tests have to be carried out on these materials to prove that with chosen stress and temperature levels satisfactory service

will be obtained without undue scaling and corrosion troubles. The manufacture of uniform forgings in the sizes required by, and exhibiting the properties guaranteed to, the turbine designers, is a hard task indeed for forgemasters.

Early turbine casings were of cast iron, but this very soon exhibited at the higher temperatures the undesirable cast-iron growth prejudicial to maintenance of fine clearances within the turbine. The temperature limit above which steel is now generally used is set at 450°F, and carbon steel castings alloyed with ½% molybdenum are used up to 900°F. Beyond that temperature molybdenum-vanadium steels are used, with additions of chromium and tungsten for the 1,050°F applications. With these additions the finished product, apart from its other characteristics, is capable of satisfactory joining by welding, a feature of great importance in simplifying the casting process. Thus the casing is cast in the form of an essential shell which is afterwards fitted by welding with the required pipes and branches. With 1,050°F machines, expensive nozzle boxes using materials highly resistant to creep are welded into the cylinder, and the double wall construction affords maximum protection in the hottest zones of the casing which itself is cast in cheaper and well-proved alloy steel. The presence of chromium is expected to combat the breakdown of carbides into free graphite which has happened on several American high-temperature plants but has not so far been met in this country.

The heat treatments applied to these forgings and castings are very carefully specified, and stress relieving after rough machining is particularly important with high temperature materials if distortion in service is to be avoided. It may consist of soaking the component for several days at a temperature which allows a relaxation of internal stresses and assists in the formation of a desirable crystalline structure of correct grain size.

At present, with 1,050°F machines, the limit of usefulness of ferritic steels is being reached, and any further advance in temperature requires the application of materials not yet fully proved. Reliability being the most important feature of a steam turbine, it is not surprising that the steelmaker and the metallurgist are sorely tried by the engineers who ask them to supply formidable amounts of data about these new materials. The data required can be summarised as follows:—

(*a*) mechanical properties at room and elevated temperatures;
(*b*) service properties of material, *i.e.* scaling resistance, corrosion resistance, cyclic heating effects associated with starts and shut-downs etc.;
(*c*) influence of steelmakers' methods of melting, deoxidation and heat treatment on the mechanical and service properties (*a*) and (*b*);
(*d*) suitability of the material to be cold worked, welded etc., both from the point of view of ease of production and reliability and stability in service;
(*e*) availability in required sizes; and
(*f*) (last but not least) the cost.

Mechanical properties of steels at high temperature are usually given in the form of percentage deformation caused in a certain time by a steady stress at a fixed temperature. Unfortunately, no satisfactory method has yet been devised of performing accelerated tests on materials to simulate creep results after 100,000 hours (about 11 years). Eight different methods of extrapolation are listed by Johnson and Tapsell[4] in their comparative research, and most of these methods were introduced over 20 years ago. Useful as they are in comparing the short time characteristics, dissatisfaction with their inconsistencies encouraged Larson & Miller[5] as recently as 1951–52 to introduce yet another method where short term tests can be used to determine long time properties. The basis of the method is the principle that for a given material a simple relationship may be established to calculate the temperature and the length of time required to run a test which would yield results equivalent to a long time test at operating temperature.

As an example from the data quoted by Larson and Miller, the following combination had equivalent breaking stresses:—

Test for 13 hours at 1,200°F = operation for 10,000 hours at 1,000°F.

It does not appear very likely that this very

[4] *Johnson, A. E., and Tapsell, H. J. 'A comparison of some carbon molybdenum steels on the basis of various creep limits.' Proc. I.Mech.E.*, 1948. *Vol.* 159.

[5] *Larson, F. R., and Miller, J. 'A time temperature relationship for rupture and creep stresses.' A.S.M.E. paper* 51-*A*-36.

simple relationship could really cover the multiplicity of factors involved in an appreciation of high temperature behaviour of steels. In fact, there is so far no reliable substitute for long-time creep tests.

Creep information required for designing steam turbine casing bolts cannot be extracted from the constant stress data previously mentioned. Special tests have to be run at a constant total deformation which, of course, is associated with load reduction or 'relaxation' as time goes on. Such tests make possible the present-day bolt designs, where reducing load is carried by the bolts between the maintenance periods (never falling below a safe minimum established by the tests). During maintenance periods these bolts are re-tightened, or if necessary heat treated and re-tightened, to make them fit for further duty.

On the whole it may be said that good mechanical properties at high temperature are associated with a coarse grain metal structure, while good mechanical properties at low temperature are associated with a fine grain structure and toughness of intercrystalline constituents.

As already mentioned, at high temperatures an improvement in creep properties is brought about by small additions of molybdenum which partly goes into a solid solution with the iron and partly affects the carbide structure. Precipitation hardening with vanadium suitably heat treated results in the vanadium carbide stiffening the grain boundaries, this addition using up some of the carbon which might otherwise have formed an iron carbide.

The problem of high temperature scaling within the turbine is very much present and has been reported quite recently to have caused a breakdown on a reheat turbine in the United States and also on an 1,135°F machine in Germany. This complicated (and perhaps not fully understood) process of forming a scale consists in oxidation, after which diffusion enables the process to continue and a scale builds up, increasing the original dimensions of the component so affected. High temperature valve spindles were attacked in this way in the two instances just mentioned, scale bridging the available clearances so that when emergency shut down was necessary some of the valves could not be moved. This experience led to the adoption of valve testing routine, in which all high temperature valves are moved throughout their working stroke at regular and frequent intervals to ensure safe operation in an emergency.

Cyclic heating of turbines associated with stopping and starting, and to a minor extent variations in initial temperature, may influence the resistance of material to creep. In some cases of austenitic materials very striking reductions in creep resistance are measured in cyclic heating laboratory tests at high temperatures (1,800°F), connected perhaps with the secondary creep process ceasing below some temperature and primary creep occurring again in each reheating. A great deal of testing is still required to establish the true position with regard to cyclic heating effects, which may be very important and are at present somewhat controversial.

Microstructural changes may occur in service in consequence of heating and cooling of turbine parts, too involved to be discussed here in detail, *e.g.* spheroidization, graphitization, carbide precipitation, sigma phase formation, temper embrittlement and recrystallization.

During shut-downs, safeguards against corrosion troubles have still to be provided. It is now generally agreed that corrosion is electro-chemical in character, moisture and oxygen playing vital parts.

The foregoing brief references to the mechanical and service properties of materials mention only some of the problems and data considered in connection with the selection of materials for modern steam turbine construction. When full information is considered in addition to all data concerning the steelmakers' methods of production, suitability of material to be cold worked and welded, availability and cost, the magnitude of the work will be appreciated. The difficulty of appreciating all this information is magnified by shortage of reliable data on many of the points mentioned. The type of information that is available sometimes by its very inconsistency suggests that perhaps not all the factors influencing the particular feature are fully recognised and isolated in the tests.

If a material is required to give better creep characteristics than can be obtained with alloyed ferritic steels, so-called austenitic materials may be selected. By suitable admixture of chromium and nickel, the γ-solid solution (or Austenite) can be maintained at

room temperatures, and Hoch's diagram (Fig. 3)[6] shows the percentages of alloying elements required to produce this kind of steel. The shaded area indicates the susceptibility of the various alloys to such phenomena as separation of carbides, sigma phase or ferrite from the solid solution. These effects have a direct bearing on the mechanical properties such as hardness, brittleness and behaviour under creep conditions.

There is no doubt that mechanical properties of austenitic materials at high temperatures are markedly superior to those of ferritic materials, and it is possible that with equivalent loading they are approximately 100°F better than the best ferritic materials. Austenitic materials, however, have some very troublesome properties. Their coefficients of linear expansion are 50% higher than those for ferritic materials and they increase, of course, with temperature. Austenitic coefficients of heat transfer also increase with increase in temperature (*i.e.* generally opposite to ferritic materials) and this combination of expansion and heat transfer properties means that thermal stresses higher than those associated with ferritic materials would have to be met with an austenitic construction. General difficulty in obtaining sufficiently large forgings and castings, as well as their cost, are also adding to the reluctance with which austenitic materials are being inserted into current steam turbine designs. At the point where the austenitic component joins the ferritic component, special attention must be given to the differential expansion coefficient of the two materials, and special pipe joints are manufactured to join austenitic and ferritic pipes by welding (Kelcalloy joint, Bardgett joint, and stabilised ferritic joint piece made by a German firm).

Welding of austenitic materials is difficult because weld deposits are subject to cracking unless duplex-type rods are employed. The ferrite in the duplex rods has high ductility at high temperature and therefore deforms without crack formation.

III.—STRESS LEVELS AND FACTORS OF SAFETY

The need to maintain the reliability of the prime mover over its service life has forced the turbine designer to consider very carefully the changes in material properties at high temperature discussed in the previous section. The old-established factors of safety based on the ultimate of the material or on yield, had to be replaced in the high temperature region by a suitable factor on deformation (permissible creep), as well as on safety from fracture.

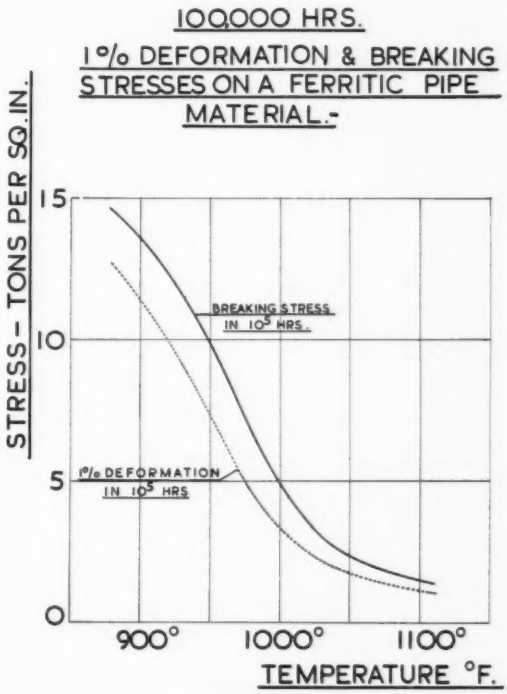


Fig. 4. Typical stress/temperature curves for a ferritic pipe material

This point will be better appreciated from Fig. 4, showing typical stress temperature curves for a ferritic pipe material. Curve (*a*) shows the stress required for 1% elongation in 10^5 hours, and curve (*b*) shows the breaking stress after 10^5 hours. If a pipe installed to operate at 1,000°F is stressed to about 3.3 tons/sq in, giving the usual 10^{-7} deformation, a 20°F increase in operating temperature would bring it to a region of failure in 10^5 hours. In selecting factors of safety on creep at high temperature, it must therefore be remembered that a practical variation in initial steam temperature may have such very far-reaching results.

[6] *Symposium: Hoch, G. 'Ein Höchsttemperatur KRAFTWERK mit einer Frischdampftemperatur von 610°C.' Z.V.D.I. 2 Aug., 1953.*

By comparison with a low temperature design, where a factor of safety on yield deformation gives about 25% higher factor on final fracture, this is a very striking change in our concepts of safety. It is therefore becoming increasingly important that the high temperature material selected is ductile at high temperature, because then measurable extensions and deformation are associated with the material approaching its complete fracture, and careful checks on deformation of high temperature parts give advance warning of impending failure. Maintenance by replacement may be deliberately designed into a machine.

In dealing with the changing character of stress levels and factors of safety, it would be a serious omission not to mention fatigue stresses. Cyclic high-frequency stresses occur in turbine parts subject to vibration and, with partial admission at inlet, high temperature blading though short may be subject to vibration. In that region, design is now based not so much on trying to avoid all possibilities of resonance, but on building sufficient strength into the blade to withstand the vibrating stresses which will occur at some harmonic of exciting frequency. High temperature properties of steel have to be allowed for there, in so far as it is possible with information which is available on endurance limits and Gerber's parabolas in that range.

At the exhaust end of the turbine where the long blades have a comparatively low frequency of vibration, the principle of design is still careful avoidance of resonant frequencies, coupled with a maximum of external damping against the build-up of dangerous amplitudes.

IV.—APPROXIMATE DISC STRESS CALCULATION

It is interesting to note that the changes in our concepts of factors of safety are not only those which have been forced upon us by the changing qualities of high temperature steel or vibration troubles in the field. 'Bursting tests of steam turbine disc wheels', reported by Robinson in 1944[7], show that elastic methods of calculating disc stresses give hidden reserves of safety when compared with the criterion of failure established in these tests. This criterion of failure is the average tangential disc stress, which can be easily derived from first principles (see Appendix):

$$S_t = \frac{2KI}{A} + \frac{F_c}{2\pi A} \quad \text{where}$$

S_t = average tengential disc stress.

K = 36.2 for steel discs at 3,000 r.p.m.

I = moment of inertia of disc section on one side of centre line (inches^4).

A = area of disc section on one side of centre line (square inches).

F_c = external loading, in pounds, due to blades and disc rim (taken from $\frac{1}{10}$ inch below root groove for the purpose of the following remarks).

Apart from the special significance given to this formula by bursting tests, it has been shown by the author that maximum elastic hub stresses can be approximately but reasonably quickly calculated on the basis of this formula. These stresses are important because large deformations are associated with a complete disc failure, and proportioning of discs is carried out on the basis of elastic stresses and on the basis of deformation permissible within the turbine.

An accurate calculation of stresses in rotating discs is lengthy and laborious. A. Stodola in his book on 'Steam & Gas Turbines'[8] gives the general equation for determining stresses in discs of variable thickness. A rotating disc of conical profile was worked out by Martin in 1923[9]. An approximate but sufficiently accurate graphical method involves replacement of the disc section by several 'equivalent' discs of uniform thickness; this was developed by Donath in 1912,[9] and translated into English by Haerle in 1918.[9] This method was further developed by Grammel, Driessen, Olsson and Held[9]. Another very close approximation for the general calculation of disc stresses, based on replacing the actual disc by a set of equivalent

[7] *Robinson, E. L. Trans. A.S.M.E. July*, 1944.

[8] *Published by Peter Smith, New York*, 1945.

[9] *See Timoshenko, S. 'Strength of materials,' Vol. II, p.* 253–4.

hyperbolic discs, is due to Arrowsmith.[10] This has been further improved and simplified within the Steam Turbine Design Department of The English Electric Company.

Where knowledge of the full distribution of disc stresses is necessary, these or similar calculations still have to be made. When dealing with preliminary turbine work it is necessary to examine many different layouts all involving lengthy disc stress calculations, and here a quick though less accurate method of calculation becomes particularly useful.

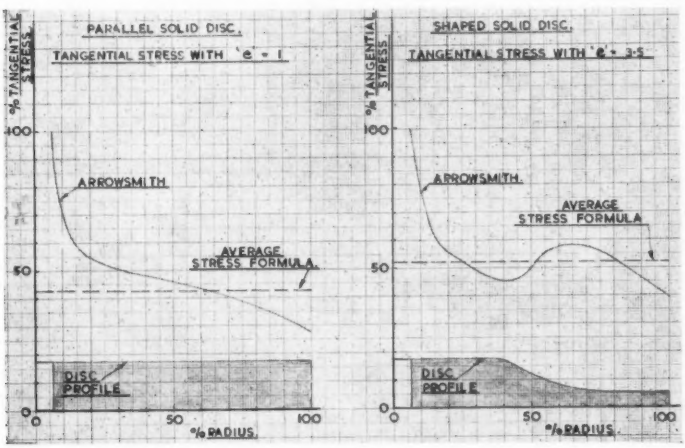


Fig. 5. Stress relationships in steam turbine discs

Such a design method has been developed by establishing an approximate relationship between the average tangential stress and the maximum stress at the hub obtained by a modified Arrowsmith's method. Many detailed disc stress calculations for discs of varying types were investigated, and by plotting the

$$\text{stress factor} = \frac{\text{maximum tangential stress at hub}}{\text{average stress}}$$

against suitable variables (Fig. 5), the most convenient method of plotting was evolved. A suitable variable finally chosen was

$$\text{shape ratio 'e'} = \frac{\text{hub width}}{\text{mean width of the disc proper}}.$$

In order to make the stress factor quickly available for a variety of typical steam turbine discs, two design curves were plotted, one applicable to solid discs and the other to separately mounted discs. The use of these curves is limited to discs with an inner bore and where the ratio of rim to bore is greater than 3.

Solid Discs

In Fig. 6 all loading above the neck of the disc was taken as external loading. If no definite neck existed, as for a parallel disc, external loading was taken from $\frac{1}{10}$ inch below the blade root groove.

This simplification, as well as the character of the method and the use of the integrator for determining the moment of inertia of disc sections ($\pm$ 5%), was mainly responsible for the $\pm$ 10% spread of results which are contained between the upper and lower envelope curves. Experience in the use of these curves soon enables much of this error to be eliminated, the general rule being that if the upper envelope curve is used, higher stresses than actual are calculated. As soon as these stresses come close to that permissible, an accurate disc stress calculation may have to be made. In practice, disc widths are often determined by the thickness of the diaphragm between the stages, and a quick check by the above method is quite satisfactory at the development stage.

Separately Mounted Discs

In general, heavier disc heads are associated with discs of this type, and if an allowance is made for self support of the disc heads, Fig. 7 could be made to coincide in practice with Fig. 6. A complicated table of self support coefficients would have to be used in such a case, thus reducing the simplicity of the method. Therefore another curve (Fig. 7) is drawn for typical separately mounted discs, where all loading above the neck is taken as external centrifugal loading.

[10] *Arrowsmith, G. 'The design of rotating discs.' 'Engineering.' 5 Oct., 1923.*

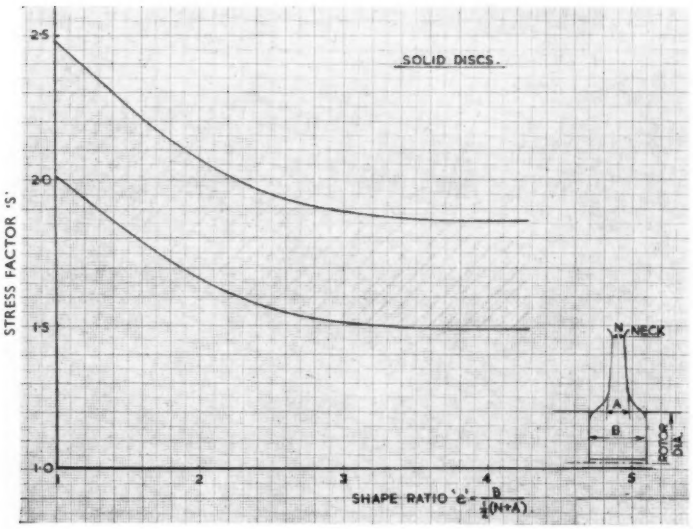


Fig. 6. Curve relating stress factor with shape of solid discs

The same remarks apply to the spread of the results and the use of the upper envelope curve, as in the case of solid discs.

Critical Radius

This important dimension can be obtained from the full stress calculation as the radius at which the degree of self support is equal to one:—

$$\frac{S_t - .3S_r}{2KR^2} = 1,$$

where S_t = tangential stress at radius R

S_r = radial stress at R

K = 36.2 for steel discs at 3,000 r.p.m.

Therefore, critical radius $= \sqrt{\left(\frac{S_t - .3S_r}{2K}\right)}$.

When altering the shape of the disc for a particular duty, it will be noted that adding width of metal below the critical radius or reducing width of metal above the critical radius reduces the maximum tangential stress. This is borne out by the falling characteristic of 'stress factor' with increasing values of 'shape ratio'.

The reason for this is that at the critical radius the tangential and radial stresses due to loading at this radius are equal to the centrifugal self loading of the disc. Above the critical radius the self loading of the disc increases rapidly, and width of metal added to the disc there reduces the load-carrying capacity of the disc. The reverse is true of adding width to the portion of the disc below the critical radius, where the self loading of the disc is relatively small and if width of metal is increased there it is mainly available for carrying external loading.

The average tangential stress in a thin ring at a radius R (see page 322) can be expressed as

$$S_t = \frac{F_c}{2\pi A} = \frac{K \times 2\pi RA \times 2R}{2\pi A} = 2KR^2$$

By comparison with a great many accurate disc stress calculations it may be said that very approximately

$$\text{critical radius} = \sqrt{\left(\frac{\text{average tangential stress}}{2K}\right)}$$

Maximum Radial Stress

It is, of course, important with a completely new disc section that some check be made on the radial stresses in the disc. By inspection of a large number of discs it was found that a maximum usually occurred either at a radius 'r' a little above the hub, or at the neck of the disc (Fig. 8).

$$\text{Approx. radial stress at 'r'} = \frac{\text{external blade loading}}{2\pi r \times \text{disc width at 'r'}} + \frac{0.7 \times \text{load due to disc above radius 'r'}}{2\pi r \times \text{disc width at 'r'}}$$

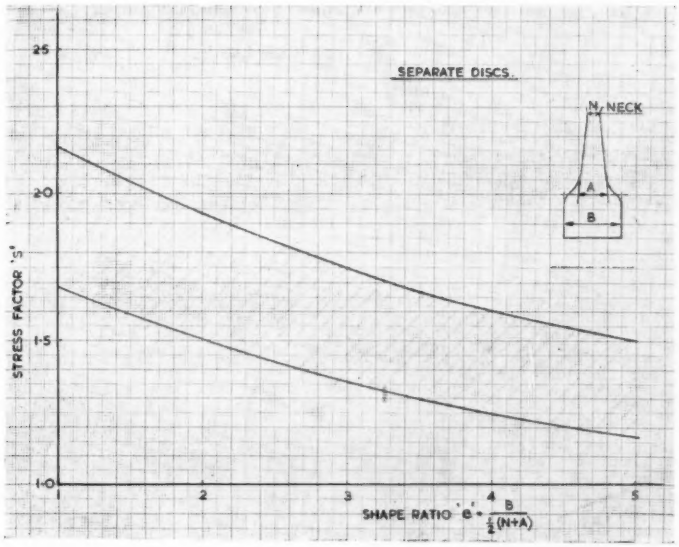


Fig. 7. Curve relating stress factor with shape of separate discs

Approx. radial stress at neck

$$= \frac{\text{external blade loading}}{2\pi \times \text{radius at neck} \times \text{disc width at neck}}$$

The use of this short calculation provides a rough check of the maximum disc stresses. Approximately a third of the time is required by comparison with a full disc stress calculation if an integrator is available or use is made of a simple 'home made' integrating grid described by Hodge in 'The Engineer' of 1946.[11]

V.—CURRENT TURBINE DESIGNS AND NEW PROJECTS

On reference to the opening paragraphs of this article it will be clear that even within the broad efficiency levels fixed by the size of the turbine and its cycle, there may yet be a demand for high and medium efficiency machines within each 'standard' group, depending on the cost of coal, load factor, cooling water temperature, etc.

This demand has been met by The English Electric Company by developing the following two 'standard' 30 MW machines.

Standand 30MW Turbines (600 p.s.i.g., 850°F)

(a) *Medium Efficiency*

This is a 2-cylinder machine (Fig. 9), impulse H.P. and double-flow reaction L.P. Both the H.P. and L.P. rotors are gashed from the solid and thus in the higher temperature range the problems of creep and expansion affecting disc shrinkage are avoided.

There are two bearings to each rotor and flexible couplings are used throughout, so that axial adjustment of rotors to the casing may be made during operation.

A single stop valve is separately anchored and connected by flex-

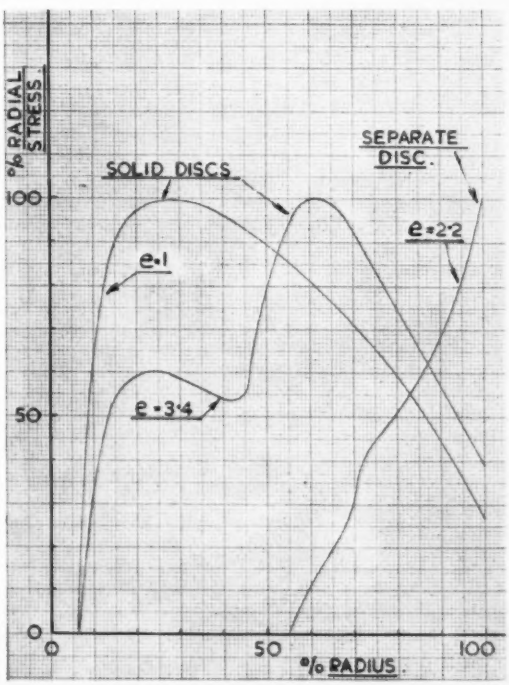


Fig. 8. Typical radial stress curves

[11] *Hodge, J. 'Stresses in turbine blading.' 'The Engineer.' 20 Dec., 1946.*

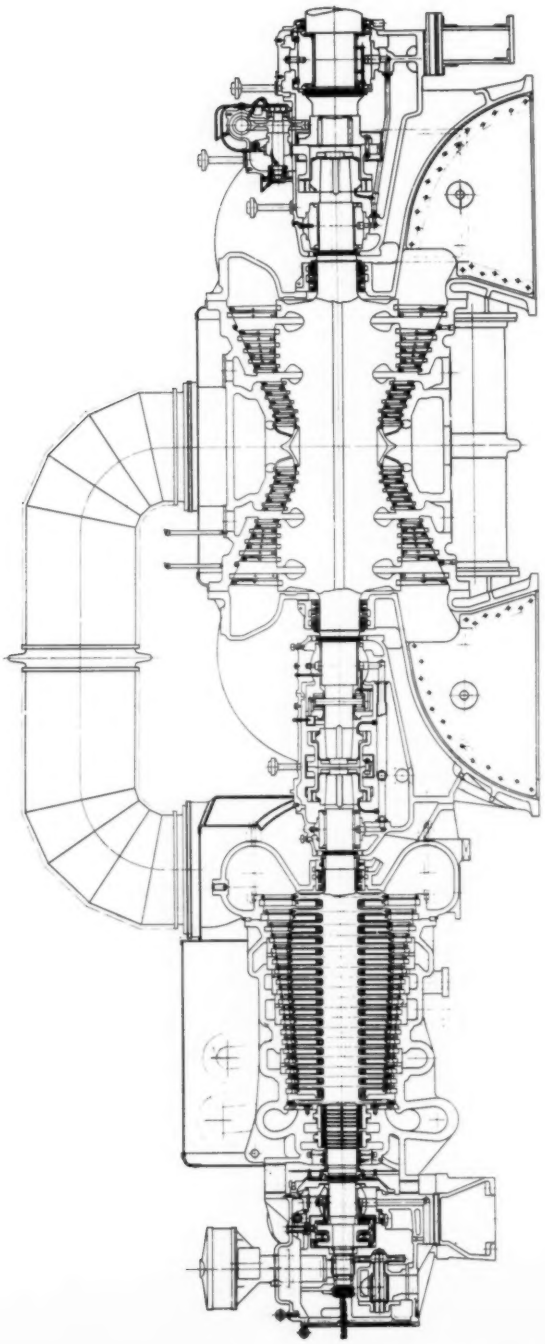


Fig. 9. Standard 30 MW medium efficiency turbine

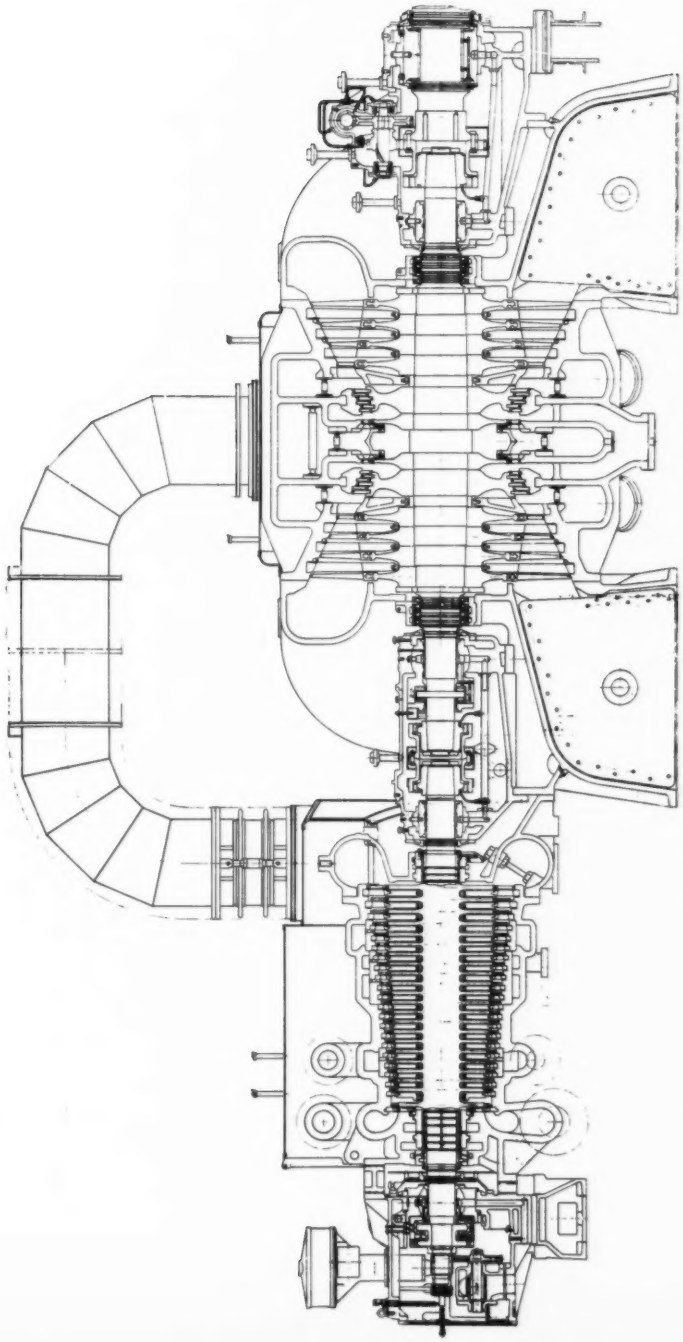


Fig. 10. Standard 30 MW high efficiency turbine

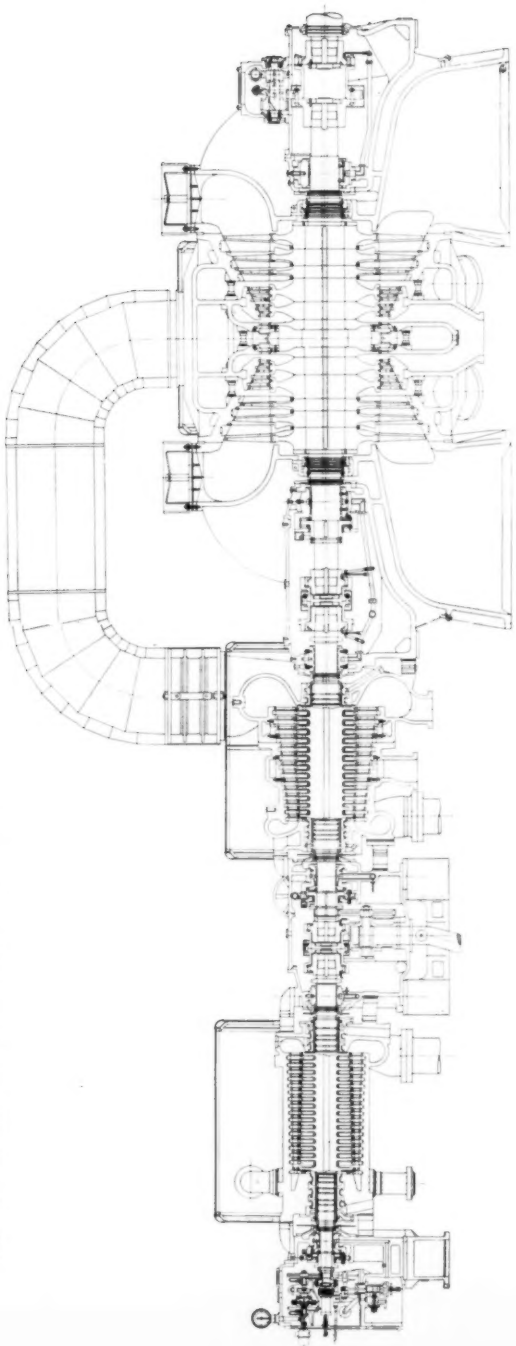


Fig. 11. *Standard* 60 *MW* 3-*cylinder turbine*

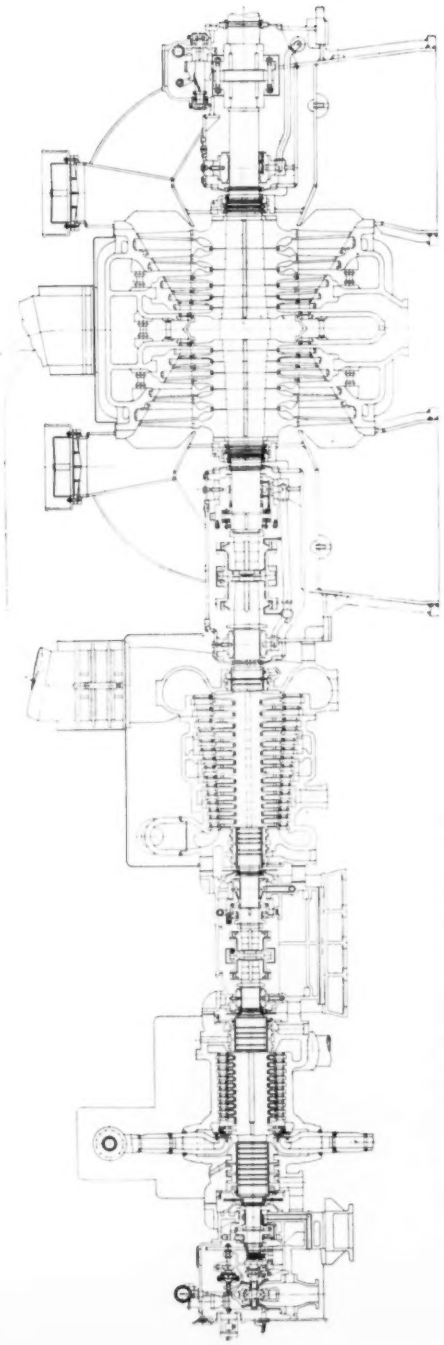


Fig. 12. *Standard* 100 *MW* 3-*cylinder turbine*

ible pipes to a chest which is bolted on to the H.P. casing.

Staging and linear speeds of blading are such as to give a medium efficiency performance.

(b) *High Efficiency*

This is also a 2-cylinder machine (Fig. 10), impulse H.P. and double-flow reaction L.P. Here the H.P. rotor is gashed from the solid but the L.P. rotor is built up of large diameter discs of high quality material shrunk on to the spindle.

Other mechanical details of construction of this turbine are broadly similar to those of the medium efficiency machine (*a*).

The carefully designed extra staging at large diameters, however, gives this turbine an excellent performance. With expensive fuel and reasonably good load factors the commercial value of such good performance is very high and much in excess of the additional cost of this machine to the power station.

Standard 60 MW Turbine (900 p.s.i.g., 900°F)

For this duty only very high efficiency 3-cylinder machines (Fig. 11) have so far been constructed by this Company, with impulse H.P. and I.P. and reaction 2-flow L.P.

The H.P. and I.P. rotors are gashed from the solid, and the L.P. rotor is built up of separate wheels shrunk on to the spindle.

Each rotor is separately supported in two bearings and transmits power through flexible couplings so that axial adjustment of rotors to casings may be made easily. This facilitates quick starting and improves internal clearances during steady operation.

There are two chests separately anchored and connected to the turbine with flexible pipes.

Standard 100 MW Turbine (1,500 p.s.i.g., 1,050°F)

This 3-cylinder machine (Fig. 12) was designed to British Standard conditions and has been developed on the basis of the 60 MW turbine just described. Again very high efficiency has been achieved by suitably staging the machine which, because of its size, capitalises improved performance at a much higher rate than a smaller set.

Double casing nozzle boxes are employed to isolate the high temperature steam from the casing which is less highly alloyed than the special nozzle boxes.

Trend of Development

These machines represent the trend of construction today and it is interesting to note that there is a large measure of agreement between the British manufacturers regarding the use of double outwards flow L.P. cylinders, mostly of built-up construction.

There is an even division of three British manufacturers favouring 3-cylinder designs and three in favour of 2-cylinder designs for the standard 900 p.s.i.g., 900°F 60 MW machines. Better efficiency as well as some improvement in reliability is obtained with three cylinders, while shorter machines are obtained with two cylinders.

Present day designs prove the soundness of the impulse-reaction principle. Most reaction turbine manufacturers use at least a single impulse stage at the beginning of expansion; all impulse turbine manufacturers use reaction in their L.P. blading design. 'English Electric' distinctive impulse-reaction designs were developed in the early 'twenties.

Today, the need for efficiencies higher than those associated with 1,500 p.s.i.g., 1,050°F steam plant can be met by re-superheating. We are once again at a point in the steam cycle development when the maximum steam temperatures are limited by the available materials, and therefore very considerable developments in the use of the reheat cycle have taken place in the U.S.A., in Britain and on the Continent.

Experience gained with the early reheat machines has led to improvements in operation and design, and little, if any, operational trouble is now associated with steam turbines using the reheat cycle. Over 50% of the plant on order in the U.S.A. today is very large reheat plant, and the general indications are that the benefits of reheat (4% to 5% on heat rate) very decidedly outweigh the additional complications. Double reheat has also been considered, usually with pressures of the order of 2,400 p.s.i.g. and above, a gain of about 1½% to 2% being associated with this further refinement. A recently described extension to the Philo Station in the U.S.A. will consist of a 120 MW turbine designed for 4,500 p.s.i.g. and 1,150°F, reheating to 1,050°F

Fig. 12. Standard 100 MW 3-cylinder turbine

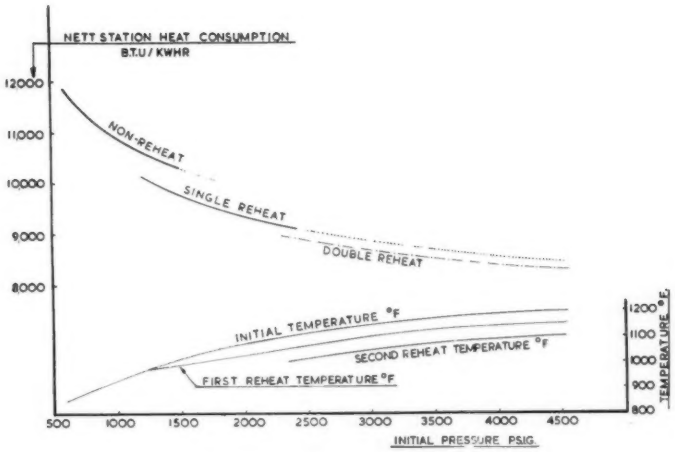


Fig. 13. *The trend in heat consumptions (based on American published figures)*

and 1,000°F in the first and second stage respectively. Calculated overall thermal efficiencies of this plant are over 40%. The pressure is well above critical, and 'once-through' type boilers are being used. The Rugby Works of The English Electric Company has a connection with the early development of this type of boiler, one of which was installed there in 1922 to supply steam to an H.P. experimental turbine.

Another interesting possibility in connection with reheat, and particularly with multiple reheat, is the use of liquid metals for transmitting heat from the boilers to reheaters situated near the turbine.

Fig. 13 illustrates the trend in heat consumptions associated with present day designs.

Feed heaters, condensers, deaerators and feed pumps are intimately connected with the operation and efficiency levels finally presented in the form of heat consumption figures. Detailed consideration of this auxiliary plant is outside the scope of this article, but the main trends of present day practice may be of interest. High level deaeration has been gaining acceptance in this country with above-atmospheric pressures maintained down to about half load. Unit evaporators have in many instances been supplanted by central evaporators, and de-mineralisation has proved itself to be both reliable and competitive in first cost. Feed pumps are now motor driven; the highest boiler pressures are usually generated by two pumps in series, reducing onerous pump duties and making easier the design of high pressure feed-water heaters.

In an attempt to reduce first cost, partly and completely outdoor steam stations have been built in the U.S.A., some of the savings in building being spent on weather-proofing the unprotected plant. The first station of this kind is now being built in Britain, but in our climate only the backs of the boilers are left exposed. Considerable economies have been obtained in conventional steam power station buildings by the use of 'unit' (one boiler per turbine) plant, which saves steam interconnection costs and reduces switchgear costs at the electrical end. Operating costs have also been reduced by centralisation of controls and by supervisory instruments which permit safe operation of modern large units with limited personnel.

New Projects

Some special steam turbine applications have recently gained a good deal of publicity; among them are geothermal projects, the Abidjan deep sea project and nuclear energy projects.

Over 200,000 kW of power is generated by the Larderello Company in Italy with steam available naturally from boreholes into volcanic strata. Large volumes of steam at about 60 p.s.i.g., 350°F are expanded to high vacua, and valuable chemicals are also extracted from the condensate.

The Abidjan (French West Africa) turbine will take advantage of the 46°F temperature difference between sea water at the surface and at the sea-bed. Large volumes of steam will be evaporated under partial vacuum and expanded through a single-stage vertical shaft turbine to a high vacuum maintained with cooling water from the deep sea level; 7.5 MW of power will be generated.

Nuclear energy reactors are the most promising sources of energy available in the near future. Ten to fifty tons of uranium per year could

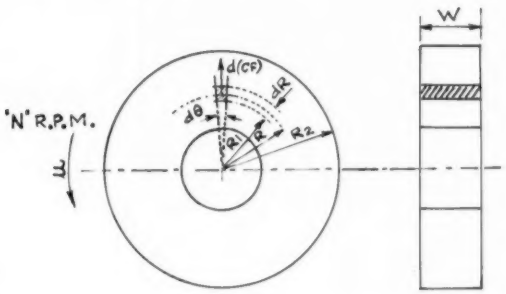
supply the present power requirements of this country, if utilised in a type of plant more advanced than that at Harwell and Chalk River. Fuel elements would operate at about 800°F and steam would be generated in heat exchangers at say 200 p.s.i.g. and 400°F. With the dwindling supplies of coal and oil, nuclear power developments will probably be very prominent during the next 15 to 20 years. The cost of these developments is not at present very competitive, chiefly due to the high costs of the nuclear portion of the plant. From the point of view of the steam turbines required for these projects, no particular construction difficulties should be encountered.

APPENDIX

Derivation of the Average Stress Formula

The following is the derivation of the formula for average tangential disc stress used in Section IV (Approximate Disc Stress Calculation) on page 322.

Referring to the diagram,



$$d(\text{c.f.}) = dm\,\frac{u^2}{R}$$

where

$$dm = \frac{w \times W \times Rd\theta \times dR}{g}, \quad w = \text{density},$$

and

$$u = \frac{2\pi RN}{60}.$$

$$d(\text{c.f.}) = \frac{\pi^2 N^2}{1800g} \cdot 2WR^2 dRd\theta$$

$$= 2KWR^2 dRd\theta, \text{ where } K = \frac{\pi^2 N^2}{1800g}.$$

$$(\text{c.f.}) = 2KW \int_{R_1}^{R_2} \int_{O}^{2\pi} R^2 dRd\theta$$

$$= 4\pi KW \frac{R_2^3 - R_1^3}{3} = 4\pi KI \quad \ldots\ldots\ldots (1)$$

where I = m. of i. of section on one side of centre-line.

Per unit angle $\frac{(\text{c.f.})}{2\pi}$, the tangential component of which

$$= \int_{O}^{\pi} \frac{(\text{c.f.}) \sin\theta}{2\pi}\, d\theta = \frac{(\text{c.f.})}{\pi},$$

if area of section carrying the stress = A, then tangential force stress

$$= \frac{(\text{c.f.})}{2\pi A} \quad \ldots\ldots\ldots (2)$$

Thus, due to the disc rotating at N r.p.m. the average tangential stress

$$S_t = \frac{4\pi KI}{2\pi A} = \frac{2KI}{A} \quad \ldots\ldots\ldots (3)$$

Due to external loading F_c,

$$S_t = \frac{F_c}{2\pi A} \quad \ldots\ldots\ldots (4)$$

Adding (3) and (4),

Average tangential stress

$$S_t = \frac{2KI}{A} + \frac{F_c}{2\pi A}.$$

K* = constant, proportional to (*a*) material density, (*b*) square of speed.

I = m. of i. of disc section on one side of centre-line (inches^4).

A = area of disc section on one side of centre-line (square inches).

F_c = c.f. due to external loading (pounds).

* For steel discs at 3,000 r.p.m. and w = .283 lb/cu in, K = 36.2.

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